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COMPOUND LOCOMOTIVES

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BY

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SECOND EDITION, REVISED AND ENLARGED

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CHICAGO
THE RAILWAY AGE AND NORTHWESTERN RAILROADER
1893

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PREFACE TO FIRST EDITION.

In the preparation of the series of articles which are here collected in book form, the aim of the author was to combine the description of the various forms of compound locomotives which have been actually used, with so much of the theory of the design of compound engines as would seem to be directly applicable to locomotive practice.

An effort has been made to present an unprejudiced analysis of each type, and to point out such advantages and disadvantages as are apparently clearly demonstrable, while carefully avoiding matters of individual preference.

Free use has been made of all available material, and the authority for data is in general given in the text. The author wishes to specially acknowledge his indebtedness to *Engineering*, and to Mr. Anatole Mallet, civil engineer, Paris; Mr. A. von Borries, locomotive superintendent of the Hanover Railroad; Messrs. Henry and Baudry, of the Paris, Lyons & Mediterranean Railway, and Mr. G. Du Bousquet, of the Northern Railway of France, for courteously supplying him with information concerning their designs.

CHAMPAIGN, Illinois, January, 1891.

PREFACE TO SECOND EDITION.

In the preparation of the second edition of this book the aim has been to add all important developments since the first edition, and to describe not so much the plans of various inventors, as to place before the reader the actual construction and practical value of compound locomotives that have been built and put into service, and to that end proposed designs have been omitted.

Extended theoretical discussion has been avoided because of the small practical value of such analysis with the limited data from actual service that is available at this time.

There has been added further consideration of the more important functions of compound locomotives, based on analyses of data and indicator cards which were not available for the first edition. Especial attention has been given to the development of such safe conclusions about the use of a compound system for locomotives as are indicated by the results of service.

Technical papers have been drawn upon to furnish illustrations for the second edition, and as it has been found impracticable to refer in each case to the publication from which the illustration was drawn, occasion is now taken to acknowledge the valuable assistance thus obtained from American and Foreign publications.

The first ten chapters have been prepared with special reference to students. Chapters XI. to XX. inclusive, refer more particularly to the different types of compound locomotives, and have been arranged for designers of locomotives. Chapters XXI. to XXIII. inclusive, are intended to place before the reader an unprejudiced comparison of the different types, and to indicate why double expansion is expected to be more economical than single expansion for locomotives.

The Appendix gives further information about the topics treated in the body of the book, and is intended for the purpose of illustration and explanation.

Valuable assistance has been given by Mr. E. M. Herr, formerly Master Mechanic of the Chicago, Milwaukee & St. Paul Railroad, and Superintendent of the Grant Locomotive Works.

DAVID LEONARD BARNES.

CHICAGO, September, 1893.

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COMPOUND LOCOMOTIVES.

CHAPTER I.

ELEMENTARY INDICATOR CARDS.

The elementary theory of steam use in compound locomotives does not differ from that of other compound non-condensing engines, but it has been found that some factors, which are of comparatively small consequence in marine or stationary work, become of importance in the locomotive. This arises largely from the wide range of power required from locomotives, and the practical necessity of keeping the valve gear and operating mechanism as free from complication as possible. The recent introduction of higher pressures and greater piston speeds in marine practice has made some of the working conditions of marine engines more nearly like the conditions of locomotive use than they have been heretofore.

The action of steam in expanding in a slow moving, elementary compound engine is well laid down in text books, and the elementary indicator cards show in a general way how steam acts in an engine. This is well understood by most of those who will be called upon to design the cylinders and valve motion of compound locomotives. Such elementary analysis is, however, of but little value as a guide to an understanding of what takes place in a compound locomotive. This results mainly from the high piston speed which causes excessive wire-drawing and compression with the valve motions ordinarily used. Such motions are universally positive and direct, and do not differ materially in action from the well-known Stephenson link, and have, generally speaking, all of its defects. Although elementary analysis has a limited application to the compound locomotive, yet it is, perhaps, best to review the elementary theory somewhat in order to properly introduce the more complicated and involved conditions, which actually exist in a practical engine.

1. Types of Compound Locomotives Commonly Used.—There are two distinct types of compound engines that have been commonly used; one has a large receiver between the cylinders, into which the h. p. cylinder exhausts, and from which the l. p. cylinder takes steam. The other form has no receiver, so-called, but may have a small space between the cylinders, consisting of the volume of the clearances of the cylinders and the volume of the space in the valve.

The first type of compound is commonly called the "receiver" type; the second, without a receiver, is generally known as the "Woolf" or "continuous expansion" type, and is only used for locomotives, in which both pistons are attached to the same crosshead. The Woolf type of expansion of steam is used in the Vauclain type, built by the Baldwin Locomotive Works, and the Johnstone type, used on the Mexican Central Railway.

2. Receiver Type of Elementary Indicator Cards.—The combined elementary indicator card from a receiver compound engine has the general form shown by Fig. 1 when no account is taken of the clearance spaces, and when it is assumed that steam is admitted and exhausted exactly at the beginning and end of the stroke, and no allowance is made for wire-drawing through the steam ports, for compression, nor for irregularity caused by the angularity of the connecting rods.

The upper part of the card, *a, b, c, d, e, f, a,* is from the h. p. cylinder, and the lower part of the card, *e, f, g, h, k, e,* is from the l. p. cylinder. The cards are on the same scale of pressures and have the same length, and are placed with respect to each other as they would be when the cranks are placed at right angles. This appears from the fact that the point *e*, the admission to the l. p. cylinder, is placed in the middle of the card from the h. p. cylinder, or just one-half a stroke later than the admission point *a* to the h. p. cylinder. The h. p. card leads to the right and the l. p. to the left, as a matter of convenience in illustration, as will appear later.

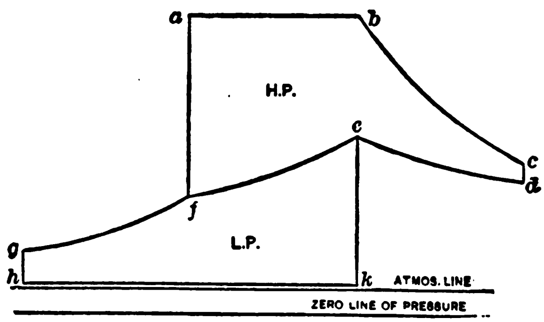


FIG. 1.
Receiver Type of Elementary Indicator Card.

The following is a description of the different lines on this combined diagram: At *a* steam is admitted to the h. p. cylinder with a pressure corresponding to the distance of *a* above the atmospheric line. Steam continues to be admitted at this pressure until the piston has advanced to the cut-off point, at half-stroke in this case, *b*. From *b* to *c* steam expands, and at *c* is exhausted into the receiver. The fall in pressure from *c* to *d* represents the drop of pressure into the receiver, and is a source of loss in compound engines, 26. The most perfect compounds have no drop of any magnitude when the h. p. cylinder opens to the receiver, 36. From *d* to *e* the h. p. piston is pushing steam into the receiver. At *e* steam is admitted to the l. p.

cylinder from the receiver, and from *e* to *f* steam is being pushed into the receiver from the h. p. cylinder, and is being taken out of the receiver by the l. p. cylinder.

The drop in pressure from *e* to *f* is the fall of pressure in the receiver, and results from the fact that the l. p. cylinder takes more steam out of the receiver from *e* to *f* than is put into it by the h. p. piston during the same time. At *f* the h. p. piston ceases to push steam into the receiver, it being at the end of the stroke. At this point also, for the purpose of illustration, it has been assumed that the l. p. valve cuts off the steam from the receiver; therefore, from *f* to *g* steam is expanding in the l. p. cylinder. The fall from *g* to *h* shows the drop in pressure at the exhaust of the l. p. cylinder to the atmosphere. From *h* to *k* is the line of back pressure in the l. p. cylinder, which is somewhat above the atmospheric line, as shown.

In all practical engines, or nearly all, the cylinders are double acting, and therefore, in the engine assumed for Fig. 1, there will be an exhaust of steam at the end of each stroke of the h. p. piston; hence, when the l. p. piston has moved to the point *f* from *e*, there will be at *f* an increase of pressure in the receiver and in the l. p. cylinder, due to the exhaust from the opposite end of the h. p. cylinder, which will cause in actual work the point *f* to rise slightly. This will appear from an examination of an actual indicator card. See Fig. 14. A different arrangement of the cut-off from that assumed for Fig. 1 would cause a somewhat different shape of combined card, but in general the description given will answer for all elementary indicator cards from receiver compounds.

3. Non-Receiver Type of Elementary Indicator Card.—In locomotive practice, so far, four-cylinder compounds without receivers are so made that the h. p. and l. p. pistons move together. This type includes the Du Bousquet non-receiver tandem, the Vauclain, and the Johnstone, of the types that have been put into practical service, and others that have been suggested but not built. The problems to be solved, when the pistons move simultaneously are, in some respects, quite different from those for receiver engines.

The Woolf, or "continuous expansion" engines, is typical of this class; the pistons move simultaneously and

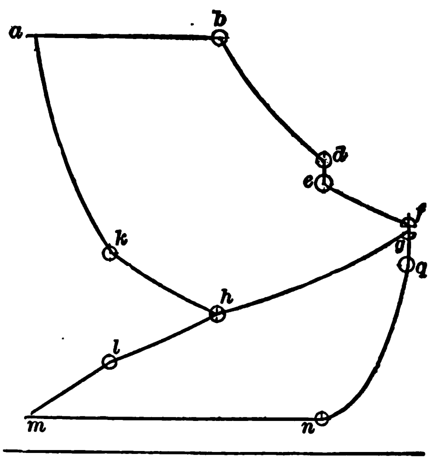


FIG. 2.
Non-Receiver Type of Elementary Indicator Card.

there is no receiver. In the simplest forms of this type, as applicable to locomotives, the h. p. and l. p. pistons are attached to the same crosshead, and the slide valves of both cylinders are operated by the same link motion. The peculiarities of the steam distribution in this arrangement of cylinders can be best examined by means of elementary indicator cards such as Fig. 2.

Referring to this figure, *a*, *b*, *d*, *e*, *f*, *g*, *h*, *k*, *a* is the h. p. card, and *g*, *h*, *l*, *m*, *n*, *q*, *g* is the l. p. card. In the h. p. cylinder cut-off takes place at *b*, and there is expansion in that cylinder until the exhaust opens at *d*. There is then a drop in pressure to *e* as the steam in the h. p. cylinder mingles with that in the passages which connect the cylinders. From *e* to *f* there is further expansion in the h. p. cylinder and the connecting passages. At *f* the l. p. steam valve opens and there is another drop in pressure to *g*.

From *g* to *h* the cylinders are in communication, and there is expansion until the l. p. steam valve closes at *h*. From *h* to *k* there is compression in the connecting passages and the h. p. cylinder, and when the h. p. exhaust closes at *k* there is further compression in that cylinder. In the l. p. cylinder the steam expands from *h* to *l*, where release occurs and the pressure drops to the ordinary back pressure line.

The fall of pressure in the l. p. cylinder up to cut-off is shown by *g h*. The pressure falls because the amount of steam pushed into the l. p. cylinder by the h. p. piston is less than the volume displaced by the l. p. piston in the same time. At the point *h* the l. p. cylinder cuts off and communication is closed between the h. p. and l. p. cylinders; hence, from *h* to *a* the steam remaining in the h. p. cylinder is compressed, for it has no outlet. This is often called "continuous expansion," as there is no pause of expansion as in the case of those engines where the steam is passed to an intermediate receiver after expansion in one cylinder.

The features of this diagram which require special attention are the losses in pressure at *d* and *f* and the compression in the h. p. cylinder. In order to prevent the drop at *d*, either the pressure in the connecting passages, valves and clearance spaces between the cylinders when the h. p. exhaust opens must be the same as that at *d*, or else the volume of the connecting passages must be practically nothing. The pressure can possibly be made the same as at *d* by adjustments of the l. p. cut-off, but it is not practicable on account of the unavoidable complications. The only feasible method of reducing this loss to an inappreciable amount appears to be to make the volume of the connecting passages very small compared with that of the h. p. cylinder. The drop in pressure at *f* can be prevented or reduced by compressing to the pressure *f* in the l. p. cylinder, or by making the l. p. clearance very small.

The question of compression in the h. p. cylinder in this type of engine is even more troublesome than in receiver engines. In order to avoid compressing to a higher pressure than the initial pressure with the usual forms of valve gear, it is necessary that the volume of the h. p. clearance space should be made large, since the pressure at *k*, where the compression caused by the exhaust closure begins, is unavoidably high. This pressure can, of course, be somewhat reduced by making the volume of the passages connecting the cylinders large, but, as has been shown, this involves a considerable drop in pressure at *d*, 37. See Figs. 11, 12 and 150.

The expedient of giving the h. p. valve inside clearance may also be employed in connection with a large clearance space to assist in keeping down the compression. In any case in which the shifting link motion is used, early cut-offs are to be avoided, both on account of this compression and to avoid the wire-drawing which results from a small port opening. The use of late cut-offs has been advocated by the builders of this class of engine for the reason just given, but that involves the wire-drawing of the steam for all light work by closing the throttle. This leads to loss of potential of pressure and is not conducive to economy, especially in compound engines, as has been shown by Professor Goss in the Purdue University shop tests. See Fig. 45. 80, 151.

It is, however, not necessary to resort to very early cut-offs in order to obtain a sufficiently great expansion, as this may be secured by using a comparatively large cylinder ratio, but at high speeds the wire-drawing and compression modifies this greatly, 77-82.

In determining the proportions for the valve gear and the size of the cylinders advisable for a tandem compound which is intended to take the place of single expansion locomotive, the most satisfactory mode of procedure will be to take actual cards from similar engines for various points of cut-off, measure the area of these cards, and finally to adjust these cards for losses or gains, according to any proposed changes in design or method of operation. An example of estimating from elementary indicator cards is given in Appendix J.

CHAPTER II.

CLEARANCE, COMPRESSION, AND CONSTRUCTION OF THE EXPANSION CURVE.

4. Clearance.—The volume included between the piston, when at the end of a stroke, and the valve face at that end is called the "clearance." It includes the volume of the steam port, the space between the piston and the cylinder head, and any other spaces that are in communication with

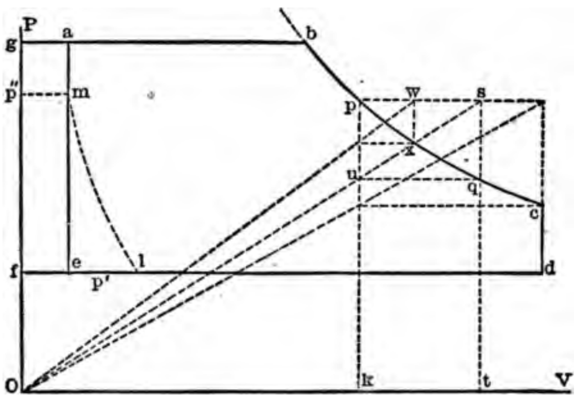


FIG. 3.
Construction of Expansion Curve.

these spaces, such as indicator pipes and cylinder drains. One of the principal effects of clearance is to make the effective or actual cut-off later than the apparent; that is, the cut-off shown by the indicator card is but the "apparent" cut-off, while the "actual" cut-off is a longer one, as shown on Fig. 3, as follows:

Let *e d* represent the stroke of a piston, and assume a cut-off at one-half stroke and ten per cent. clearance. Then *a b* is one-half of *e d*, and the apparent ratio of expansion is 2. Lay off $e\ f$ equal to one-tenth of $e\ d$, then $f\ e$ or $a\ g$ represents the clearance. The volume which is filled with steam when cut-off takes place is $g\ b$, and this expands until it fills the volume of $f\ d$. The actual ratio of expansion is therefore $f\ d$ divided by $g\ b$, or as drawn in Fig. 3 it is: $\frac{1 + .1}{.5 + .1} = \frac{1.1}{.6} = 1.83$ instead of 2. Expressing this as a formula, the actual ratio of expansion is

$$\frac{1+k}{n+k},$$

in which k is the clearance expressed as a decimal of the volume displaced by the piston in one stroke, and n is the apparent cut-off, or one divided by the apparent ratio of expansion. The point c on the expansion curve is, of course, higher with a ratio of expansion of 1.83 than with a ratio of 2, and hence the mean pressure between b and c is higher. In making calculations the actual ratio of expansion should of course be used, but the formula, 7, will not then give correct results, as by it the mean pressure between g and c is found, and not that between a and c, and a correction must therefore be made which necessitates additional calculation. It is better in most cases to make use of a graphical construction. For example, see Appendix G.

5. **Construction of the Expansion Curve.**—A simple method of plotting points on the hyperbolic expansion curve is the following, which requires only a triangle and a straight edge: In Fig. 3 let $O\ V$ be the zero line of pressures, $O\ P$ the zero line of volumes, and p a known point on the hyperbola. Through p draw $p\ s$ parallel to $O\ V$, making it of any convenient length. Draw $p\ k$ and $s\ t$ perpendicular to $O\ V$ and draw $O\ s$. Through the point u where $O\ s$ crosses $p\ k$, draw $u\ q$ parallel to $O\ V$, and where this line cuts $s\ t$ at q is a second point on the curve. Any number of other points can be found from p or q in a similar manner, as indicated

in Fig. 3. An advantage of this method is that the distance of a point from $O P$ can be selected at pleasure, as it will be always directly under the point to which the diagonal is drawn, as q and s, or x and w, 41, 43.

6. **Compression.**—Compression or cushioning in compound locomotives is a factor of steam distribution which it is more difficult to dispose of satisfactorily than in single expansion engines. For economy of steam, the pressure in the clearance space, when the steam valve opens, should not be far from, but somewhat less than, the initial pressure, while the necessary pressure for "cushioning" the reciprocating parts is a problem in itself, and is generally regulated by the lead of the valves.

In a single expansion engine having an initial pressure of 175 pounds absolute, and a back pressure of 18 pounds absolute, it is possible to compress to 9.7 times the back pressure before the initial pressure will be exceeded. But in a compound, if the receiver pressure is 70 pounds absolute, the possible range of compression is for the h. p. cylinder from 70 to 175 pounds, and for the l. p. cylinder from 18 to 70 pounds, or 2.5 times in the former, and about 3.9 times in the latter. It will be at once apparent that the valve adjustment for compression in the compound is a much more difficult problem than in the single expansion engine.

For example, with 5 per cent. clearance in a compound and the pressures as just stated, the pressure in the clearance space at the end of the stroke would equal the initial pressure in the h. p. cylinder when the exhaust closed at $2.5 \times .05 - .05 = .075$ of the stroke from the end, or at 92.5 per cent. of the stroke, as it is frequently stated. In the l. p. cylinder, an exhaust closure at 85.5 per cent. would fill the clearance space with steam at receiver pressure. With 10 per cent. clearance, and the same pressures as before, the earliest allowable points of exhaust closure would be 85 per cent. in the h. p. and 71 per cent. in the l. p. cylinder. It is practically impossible to get such late exhaust closures at early cut-offs with a link motion, 73–81.

It will be seen from this that a large percentage of clearance in a compound engine will reduce compression

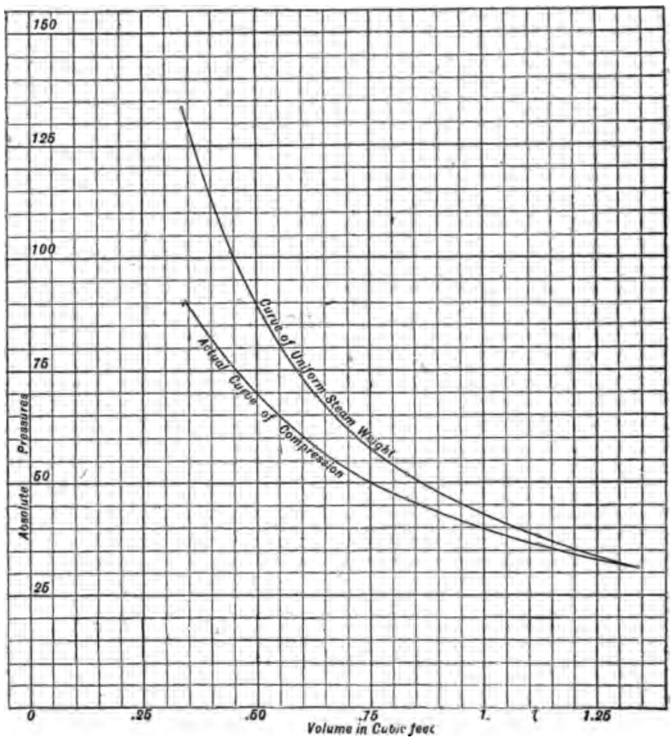


FIG. 4.
Actual Curve of Compression.

and may be a positive advantage, so far as the distribution of power between the cylinders is concerned, also large clearance spaces assist in the reduction of high compression at fast speeds.

An approximation to the relations between the back pressure, the pressure from compression, the point of exhaust closure and the clearance, can be expressed in a general formula as follows: Referring to Fig. 3, let p' represent the back pressure and p'' the pressure in the clearance space at the end of the compression, both measured from the zero line of pressures; let l be the point of exhaust closure, $l m$ the compression curve which is considered as a rectangular hyperbola, $d e$ the stroke of the piston, and $f e$ equal k, the clearance as before. Then the fraction of the stroke at which the exhaust should close to produce p'' is:

$$\frac{d\,l}{d\,e} = 1 - \left(\frac{p''}{p'} - 1\right)k.$$

It should be remembered that this formula is but an approximation, as the real compression curve is not a rectangular hyperbola, but has more nearly the form of the lower curve in Fig. 4. This modification of the compression curve is produced by the cooling action of the walls of the cylinder, the face of the piston, and the walls of the steam passages, all of which have to be heated to the temperature of the steam which rises during compression. This difference between actual and hyperbolic curves, in Fig. 4, indicates a loss due to clearance. Clearance compels compression, and compression carries with it this type of loss.

The problem of determining the amount of compression necessary to cushion the reciprocating parts does not differ essentially in compound and single expansion engines, except that with compounds the weight of the reciprocating parts is necessarily greater.

To further illustrate the difference between the actual curve of compression, and the hyperbolas drawn from any point in that curve, and to show the decrease of steam weight during compression, reference is made to Figs. 5 and 6, which show some actual indicator cards taken from a locomotive. The actual clearance in the engine is 8 per cent., and is represented by the full vertical lines. The

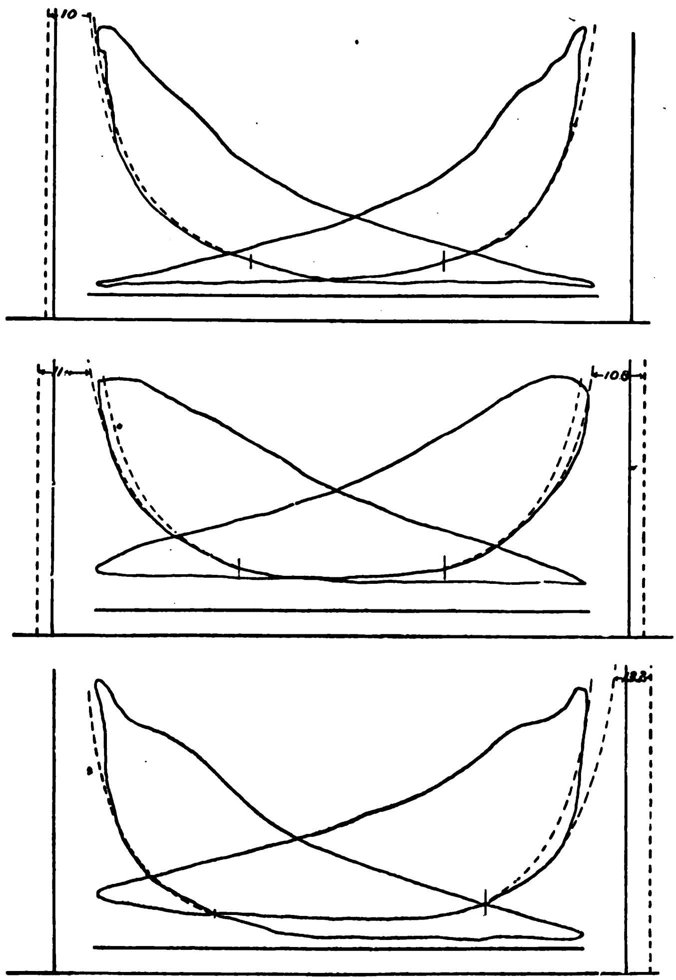


FIG. 5.
Difference between Actual Curve of Compression and Hyperbola.

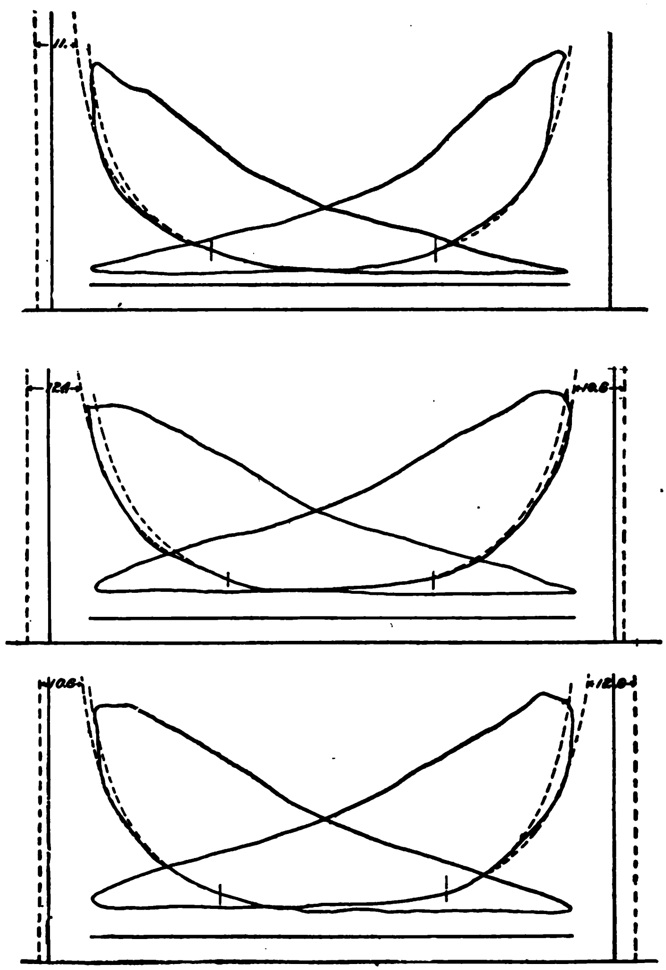


FIG. 6.
Difference between Actual Curve of Compression and Hyperbola.

dotted lines for comparison with the curve of compression are hyperbolas, one of which is drawn from a point of the compression curve after the exhaust valve is closed, and is based on the actual clearance. This dotted line is always the one which falls inside of the compression curve. The other dotted line is an hyperbola that is drawn to approximate closely to the actual curve of compression. This second line is drawn from the same point of the actual expansion curve as the first dotted line, and the clearance which would give this hyperbola is shown by the dotted vertical line. This would indicate that an approximation to the actual curve of compression may be made by assuming an hyperbola for the shape of the curve of compression, and changing the clearance to suit; that is to say, the actual compression curve approximates to an hyperbola based on a greater clearance than is actually used in the engine from which the cards were taken. The amount of this greater clearance is given in the illustrations.

These comparative lines on Figs. 5 and 6 are hyperbolas, and therefore show less decrease in weight of steam during compression than would be shown if the curve of equal steam weight had been used for comparison, as is evident from Fig. 23a.

CHAPTER III.

MEAN EFFECTIVE PRESSURE.

7. Formula for Calculating Mean Effective Pressure.— For calculating the pressures at the various points of elementary cards, we can without serious error make use of the ordinary formulas, and assume that pressures of steam vary inversely as the volumes, the curves of expansion and compression then being rectangular hyperbolas. On this basis, the absolute mean pressures for such lines as *a b c*, Fig. 3, are determined by the formula: 43.

$$p = P\frac{1 + \text{hyp. log. r}}{\text{r.}}$$

This will be recognized as the ordinary formula for mean pressures, and in which P is the absolute initial pressure, r is the ratio of expansion, *i. e.*, volume at cut-off divided by volume at end of stroke or at exhaust, as the case may be, and p is the absolute mean forward pressure. The absolute pressure is the gauge pressure plus the atmospheric pressure, which is practically 14.7 pounds per square inch. The term " hyperbolic " as applied to logarithms refers to the " Natural " or " Naperian " logarithm. An example of the application of the above formula will be found in Appendix A. This formula is applicable to such lines of the card as *a b c* when *a b* is parallel to the atmospheric line, as it is practically in engines supplied from a boiler and working at slow speeds. For calculating the mean pressure between *b* and *c*, *d* and *e*, *e* and *f*, or for other expansions or compressions in which the part of the card considered is wholly within the hyperbola, and where the line of constant pressure as *a b* is not included, the following formula is to be used:

$$p = P\frac{\text{hyp. log. r}}{\text{r}-1}$$

For example see Appendix B and Appendix F.

8. **Difference Between Calculated and Actual Mean Effective Pressure.**—The foregoing method serves to illustrate what the action of steam in locomotive cylinders is frequently assumed to be, and is worth perusal by the student; but for actual practice, the mean effective pressure in either cylinder differs so much from that given by

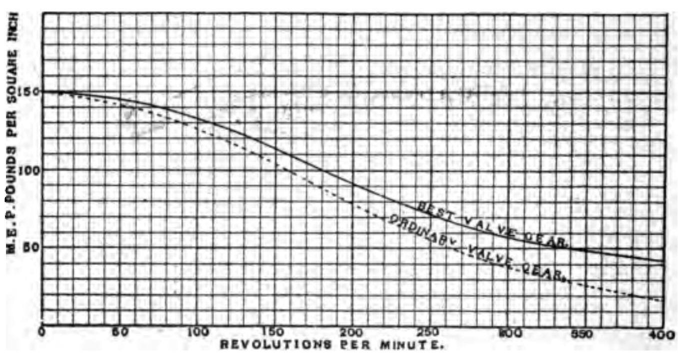


FIG. 7.
Reduction of M. E. P. as Speed Increases.

calculation, that the only safe course to pursue is to draw the preliminary indicator cards by modifying actual cards from practice, as is explained further on.

As a more forcible illustration of this difference, Tables B, C, D, E, F, G, and H, have been prepared from the actual indicator cards Figs. 14 and 15, taken from a Schenectady ten-wheel two-cylinder receiver compound on the Central Pacific Railroad. Columns I, K and L show how wide is the variation between the calculated and actual mean effective pressures when the calculations are based on the elementary indicator cards. Reference to these tables is also made under the head of "Cylinder Ratios," chapter VII.

9. **Decrease of Mean Effective Pressure as Speed Increases.**— Fig. 7 shows the decrease, in a single expansion engine, of the maximum mean effective pressure per square inch of piston, with the best and the ordinary valve gears,

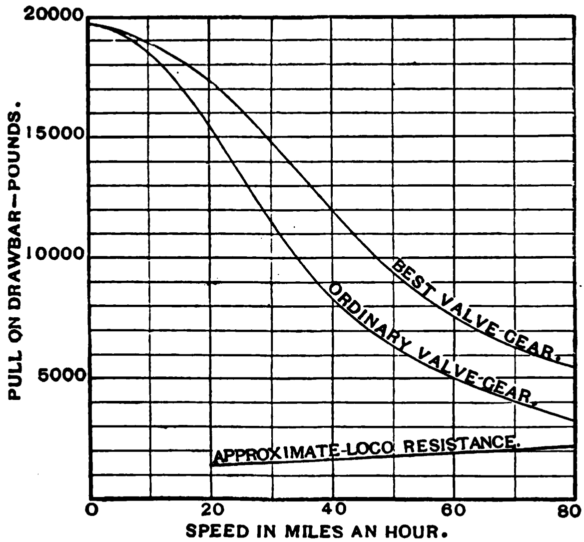


FIG. 8.

Reduction of Power as Speed Increases.

which follows an increase in the number of revolutions per minute of locomotive driving wheels. Boiler pressure, 175 pounds per square inch absolute. This shows the need of careful attention to valve gear dimensions, 77–82.

10. **Effect on Draw Bar Pull of Decrease of Mean Effective Pressure as Speed Increases.**— Fig. 8 shows the decrease in the maximum pull on draw bar of a single expansion engine which follows an increase in speed of a 19×24 locomotive with 5½ foot driving wheels, with the best valve gear and with the ordinary valve gear.

11. Increase of Per Cent. of Total Power Consumed by Locomotives and Tenders which follows a Decrease of Mean Effective Pressure Due to Speed.— Fig. 9 shows how the per cent. of total power generated by the cylinders and consumed by the locomotive and tender together, increases as the speed increases, regardless of any change there

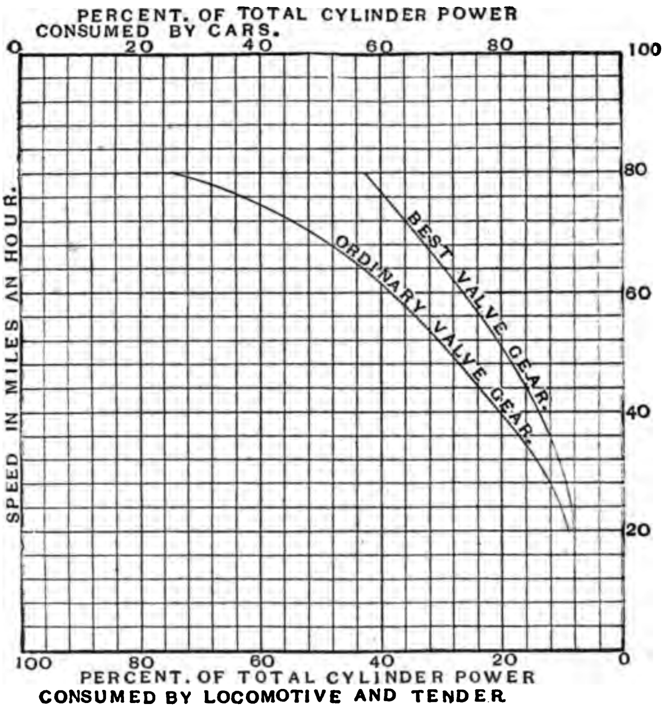


FIG. 9.

Per cent. of Power Consumed by Locomotive at Various Speeds.

may be in train resistance. This is readily deduced from Fig. 8 by comparing the total draw bar pull with the approximate locomotive resistance.

It is clear from these diagrams that at high speeds almost the entire power of the locomotive cylinders is consumed by the locomotive and tender, not because the head air resistance, or the locomotive and tender resistance, increases greatly, but almost solely because of the decrease of mean effective pressure in the cylinders brought about by wire-drawing, compression and early cut-off at high speeds. The worse the design of valve motion and steam passages, the sharper will be the inclination of the curve in Fig. 9 to the left. A misunderstanding of the real condition on the part of some writers has led to the conclusion that this inclination is due to a great increase in head air resistance The fallacy of such a conclusion appears at once from an examination of Figs. 7, 8 and 9.

Fig. 10 shows the advantage of using a large driving

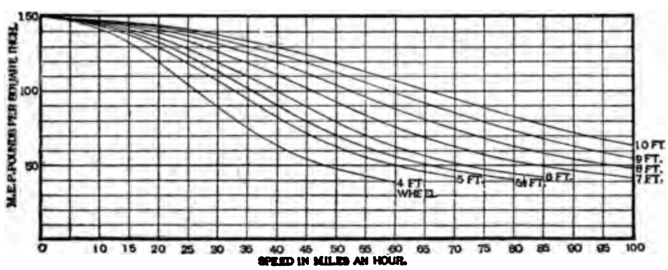


FIG. 10.

Effect of Large Drivers on M. E. P. at High Speed.

wheel on a locomotive. All that this diagram, Fig. 10, shows, applies with greater force to compounds, as the loss in power with compounds increases more rapidly as the speed increases than with single expansion engines. The mean effective pressure given in Fig. 10 is that which will be obtained when the steam valves are controlled by the best types of valve motion now used, and when the boiler pressure is 160 pounds per square inch by gauge.

Figs. 11 and 12 show very clearly how the mean effective pressure is reduced as the speed increases in a Vauclain compound. These cards, Nos. 1 to 13, were taken from a ten-wheel freight engine on the Chicago, Milwaukee

& St. Paul road. Table A gives the data calculated from these cards, and Fig. No. 13 is a diagram showing the decrease of mean effective pressure as the revolutions per minute increase. These cards are intended to illustrate

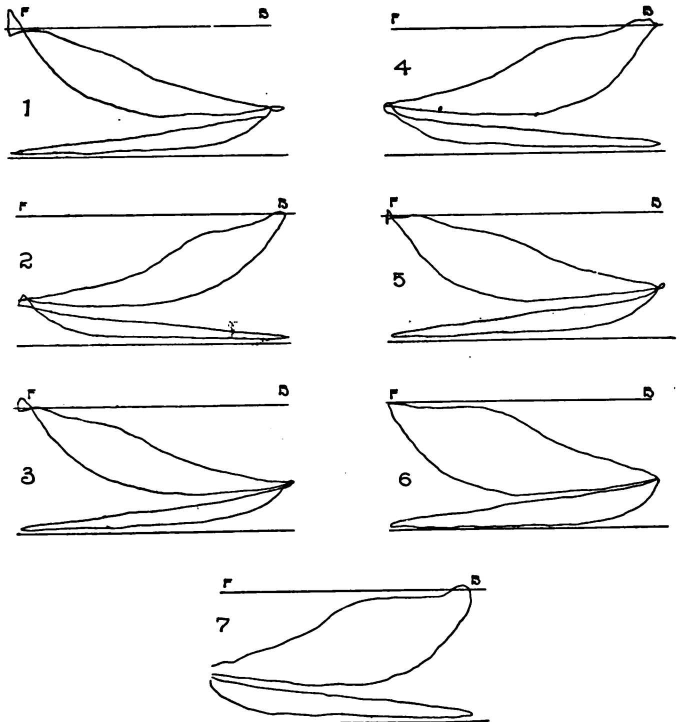


FIG. 11.
Actual Indicator Cards Showing Decrease of M. E. P. as Speed Increases.

what takes place in any engine, compound or single expansion, as the speed increases, and shows how the hauling power of a freight engine decreases as the speed increases. Card No. 1 shows, perhaps, more clearly than any of the others how compression and wire-drawing robs the engine of its power at high speed. From this it is clear that if a locomotive is proportioned so that its cylinder power

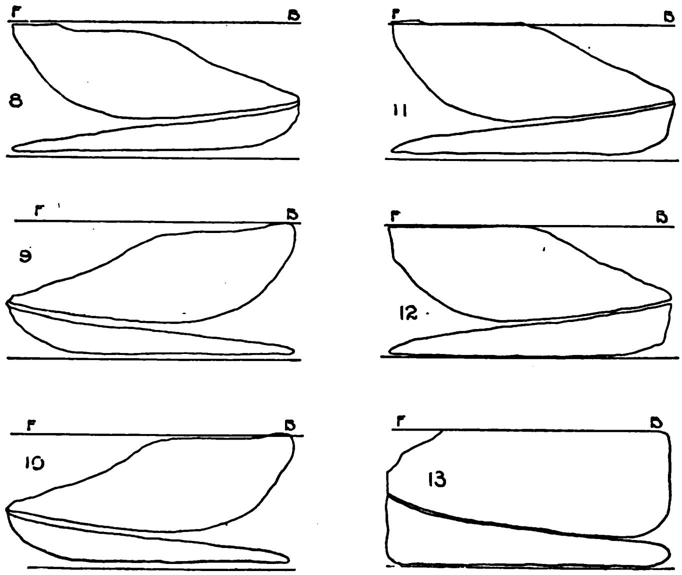


FIG. 12.
Actual Indicator Cards Showing Decrease of M. E. P. as Speed Increases.

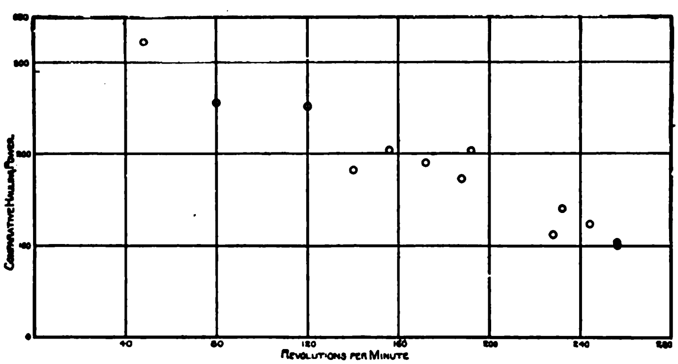


FIG. 13.
Diagram Showing Decrease of Hauling Power as Speed Increases.

at low speed is just about sufficient to slip the wheels, it will have far too little cylinder power to slip the wheels at high speed. This then is an illustration of the need of an increase of cylinder power to haul heavier trains at high speeds, and it is evident that the simplest and best way to increase the cylinder power is to reduce the wire-drawing and compression.

TABLE A.

Giving Data with Reference to Indicator Cards Nos. 1 to 13, taken from a Ten-Wheel Vauclain Compound Freight Engine on the Chicago, Milwaukee and St. Paul Railroad.

No. of card. -	1	2	3	4	5	6	7	8	9	10	11	12	13
No. of reverse lever notch.	1	1	1	1½	1½	1½	2	2½	2½	2½	2½	2½	7
Cut-off h.p. cylinder, inches.	12.25	12 25	12 25	13 25	13 28	13 28	14 25	15 41	15 44	15 44	15.41	15.41	21.62
Cut-off l. p. cylinder, inches.	15 06	15 00	15 06	15 94	15 90	15.90	16.87	17 62	17 75	17.75	17.63	17.63	22.75
Revolutions per minute. - -	256	256	228	244	232	140	188	192	172	156	120	80	48
Boiler pressure, absolute - -	191	191	185	185	183	189	192	190	192	186	190	185	191
Mean effective pressure, h. p. cylinder. -	37.50	41 25	40.00	51 88	47 50	64.50	68 75	70.00	75 00	78 75	82.50	81.25	116.25
Mean effective pressure, l. p. cylinder. -	13.75	12 50	15 00	15 00	20.00	25.00	22 50	28.75	25 00	27 50	37.50	38.75	46.25
Mean effective pressure, l. p. cyl., reduced to equivalent for h. p. cyl.	40 43	34.75	44 10	41.70	58.80	73.50	62.55	84 52	69.50	76.45	110.25	113.93	128.56
Proportional No. showing comparative hauling power	1.025	1 00	1 107	1.231	1.399	1.816	1 728	2.033	1.901	2.042	2.535	2.568	3.221
Pressure at admission to h. p. cylinder.	165	175	168	178	168	173	180	170	176	171	176	168	167

CHAPTER IV.

DIFFERENCES BETWEEN ELEMENTARY AND ACTUAL INDICATOR CARDS.

12. Difference between Apparent and Actual Cut-off.—Figs. 14 and 15 show a set of actual indicator cards from a two-cylinder receiver compound of the Schenectady type on the Southern Pacific Railroad, having the following general dimensions :

Diameter of H. P. Cylinder	20	inches
" " L. P. "	29	"
Stroke of Pistons	24	"
Diameter of Drivers	69	"
Number " "	6	
Weight on "	96,680	lbs.
" of Engine, loaded	129,700	"
" " Tender, "		
Heating Surface	1736.2	sq. ft.
Grate "	29.26	"
Heating " per sq. ft. of Grate	60.7	"
Heating Surface per sq. in. Cyl. Area, L. P.	2.63	"

Outside lap of Valve, H. P.	$1\frac{1}{8}$	inches
" " " " L. P.	$1\frac{1}{8}$	"
Inside Clearance, H. P.	$\frac{5}{16}$	"
" " L. P.	$\frac{3}{16}$	"
Size of Steam Ports, H. P. Cyl.	$2\frac{1}{8}$x18	
" " " " L. P. "	$2\frac{1}{8}$x20	
" " Exhaust " H. P. "	3x18	
" " " " L. P. "	3x20	
Cyl. Area per sq. in. flue opening	1.11	sq.in
Per cent. of Weight on Drivers	74.54	
Clearance H. P. Front	1026	cu. in.
" " " Back	1178	"
" L. P. Front	1386	"
" " " Back	1220	"

Table B shows the difference between the "actual" cut-off, taking into account the clearance, and the "apparent" cut-off measured from the valve motion when the engine is out of service, and not as taken from indicator cards, and does not therefore include lost motion and springing of the parts. The difference between these is so great as to emphasize the need of always basing calculations and examinations on the actual instead of the apparent cut-off. This table also shows the effect of clearance in increasing the actual cut-off beyond the apparent cut-off.

TABLE B.

Showing the difference between the "Actual" Cut-off, counting the Clearance, and the "Apparent" Cut-off, Measured from the Valve Motion when the Engine is out of Service, and not taken from Indicator Cards.

Actual Card No.	A Revolutions per minute.	B Miles per hour.	E Piston speed in feet per minute.	C Actual cut-off including clearance. Per cent. H P	 L P	D Apparent cut-off Per cent. H P	 L P
1	30	6.16	120	86.4	87.3	84.5	86.
2	50	10.26	200	83.8	84.2	81.2	82.8
3	60	12.32	240	76.6	78.5	73.0	76.8
4	144	29.56	576	68.2	71.2	63.5	68.8
5	180	36.95	720	58.4	61.6	52.1	58.5
6	240	49.27	960	58.4	61.6	52.1	58.5
7	240	49.27	960	50.2	55.1	42.7	51.5
8	300	61.58	1200	50.2	55.1	42.7	51.5
9	330	67.74	1320	50.2	55.1	42.7	51.5

13. Difference between Actual and Elementary Mean Effective Pressures in High-Pressure Cylinder.— Table C gives the elementary or theoretical mean effective

TABLE C.

Showing the Elementary or Theoretical Mean Effective Pressure in the High-Pressure Cylinder, based on the Elementary Indicator Cards and on Boiler Pressure.

Actual Card No.	C(h.p) Actual cut-off including clearance. Per cent.	F Boiler pressure by gauge. Pounds per sq. in.	G(hp) Gauge pressure at beginning of stroke. Pounds per sq. in.	H Average receiver pressure, gauge. Pounds per sq. in.	I Theoretical mean effective pressure in h. p. cylinder, based on boiler pressure, no clearance, back pressure equal to average receiver pressure. No compression. Pounds per sq. in.	K Actual mean effective pressure in h. p. cylinder. Pounds per sq. in.	L Per cent. of theoretical mean effective pressure (I) actually obtained in h. p. cylinder.	F1 Theoretical mean effective pressure, column (I) corrected for the clearance in h. p. cylinder.
1	86.4	152	151	58	92.3	89.1	96.8	165.3
2	83.8	142	142	54	84.9	83.2	98.0	153.9
3	76.6	152	152	54	93.0	83.0	89.2	162.0
4	68.2	150	149	50	88.5	83.6	94.5	153.5
5	58.4	160	157	47	93.8	54.2	57.8	155.8
6	58.4	160	160	46	94.8	42.4	44.7	155.8
7	50.2	160	160	46	86.0	32.9	38.3	147.0
8	50.2	160	160	45	87.0	31.3	36.0	147.0
9	50.2	165	165	45	91.2	31.0	34.0	151.2

pressure in the h. p. cylinder, based on elementary indicator cards and on boiler pressure, and includes no consideration of clearance or compression, the back pressure being taken equal to the average receiver pressure. This is compared with the actual mean effective pressure, and shows how great is the reduction of power, and to some extent economy, resulting from wire-drawing and compression.

TABLE D.

Showing the Theoretical Mean Effective Pressure in the High-Pressure Cylinder, based on the Elementary Indicator Card, and with other assumptions used for Table C.

Actual Card No.	C (h. p.) Actual cut-off including clearance. Percent.	F Boiler pressure by gauge. Pounds per sq. in.	G (h. p.) Gauge pressure at beginning of stroke. Pounds per sq. in.	H Average receiver pressure, gauge. Pounds per sq. in.	J Theoretical mean effective pressure in h. p. cylinder, based on pressure at beginning of stroke, no clearance, back pressure equal to average receiver pressure. No compression. Pounds per sq. inch.	K Actual mean effective pressure in h. p. cylinder. Pounds per sq. in.	M Per cent. of theoretical mean effective pressure (J) actually obtained in h. p. cylinder.
1	86.4	152	151	58	91.3	89.1	97.8
2	83.8	142	142	54	84.9	83.2	98.0
3	76.6	152	152	54	93.0	83.0	89.3
4	68.2	150	149	50	87.5	83.6	95.6
5	58.4	160	157	47	91.1	54.2	59.5
6	58.4	160	160	46	94.8	42.4	44.7
7	50.2	160	160	46	86.0	32.9	38.3
8	50.2	160	160	45	87.0	31.3	36.0
9	50.2	165	165	45	91.2	31.0	34.0

Table D shows the theoretical mean effective pressure in the h. p. cylinder based on elementary indicator cards and on the pressure at the beginning of the stroke, and with the other assumptions used for Table C. This shows the loss in power, and to some extent economy, resulting from wire-drawing and compression. The close approximation of the results given in Tables C and D is due to the important fact that this two-cylinder compound, Figs. 14 and 15, has very large throttle valve and steam pipes,

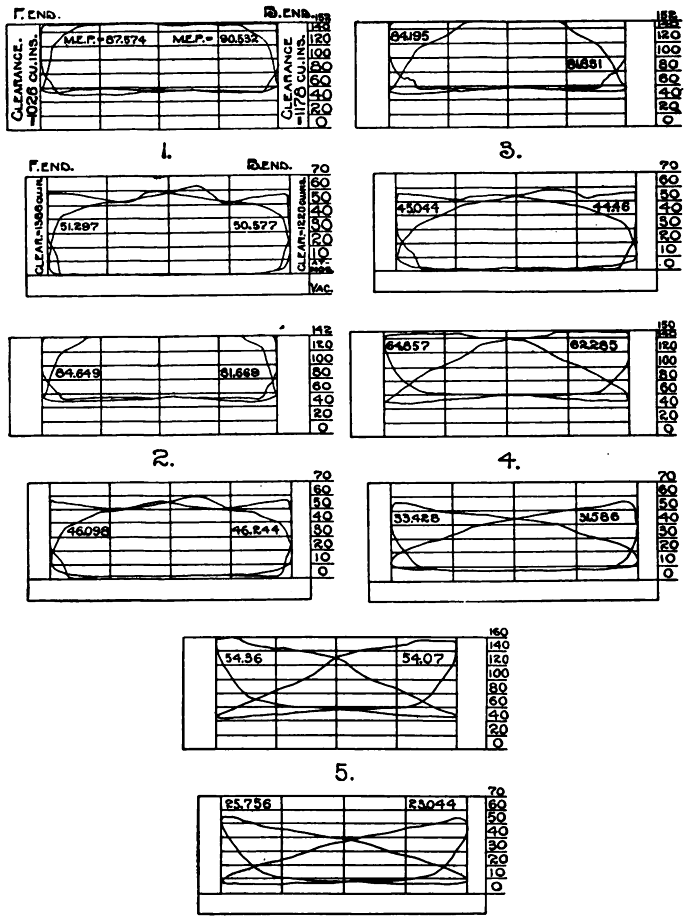


FIG. 14.
Indicator Diagrams from Two-Cylinder Receiver Compound.

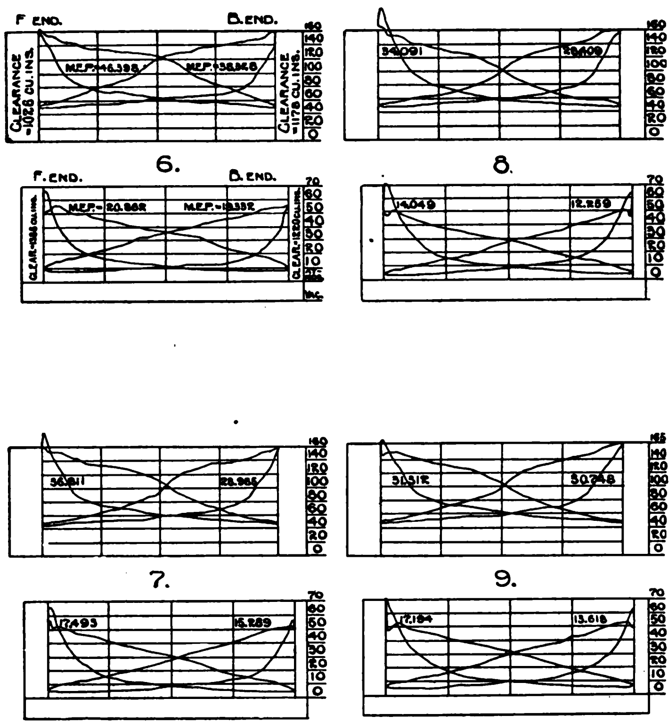


FIG. 15.

Indicator Diagrams from Two-Cylinder Receiver Compound.

and was operated with a full open throttle. There was but little, if any, loss in steam pressure between the boiler and the steam chest.

14. Differences between Actual and Elementary Mean Effective Pressures in Low-Pressure Cylinder.—Table E gives the theoretical mean effective pressure in the l. p. cylinder based on the average receiver pressure, the actual cut-off, and on 5 pounds per square inch back pressure, and the other assumption used for Tables C and D.

This shows the loss in power, and to some extent the loss in efficiency in the l. p. cylinder due to wire-drawing and compression, and shows the futility of any attempt to use the common theory of steam engines deduced from elementary indicator cards when designing compound locomotives under the ordinary conditions and with the ordinary valve gears and ports.

TABLE E.

Showing the Theoretical Mean Effective Pressure in the Low-Pressure Cylinder, based on the Average Receiver Pressure, the Actual Cut-off, on Five Pounds per Square Inch Back Pressure, and the other Assumption used for Tables C and D.

Actual Card No.	C (l. p.) Actual cut-off including clearance. Per cent.	H Average receiver pressure, gauge. Pounds per sq. in.	N Theoretical mean effective pressure for l. p. cylinder, based on average receiver pressure, and on 5 lbs. per sq. in. back pressure, and no compression. Pounds per sq. in.	O Actual mean effective pressure in l. p. cylinder. Pounds per sq. in.	P Per cent. of theoretical mean effective pressure, (N) actually obtained in l. p. cylinder.	E 1 Theoretical mean effective pressure column (N) corrected for the clearance in l.p. cylinder.
1	87.3	58	52.3	50.9	97.4	52.4
2	84.2	54	49.6	46.2	93.0	49.5
3	78.5	54	47.6	44.8	94.0	49.5
4	71.2	50	41.8	32.5	77.6	47.4
5	61.6	47	36.4	24.4	67.1	42.8
6	61.6	6	35.5	20.1	56.6	43.8
7	55.1	46	33.7	16.4	48.6	39.0
8	55.1	45	32.8	13.2	40.3	37.2
9	55.1	45	32.8	15.4	46.9	39.8

Table F gives the theoretical mean effective pressure in the l. p. cylinder, based on the admission pressure, and with the other assumption as given for Table E. This table also shows the loss in power and to some extent the loss in efficiency, resulting from compression and wire-drawing in the l. p. cylinder.

TABLE F.

Showing the Theoretical Mean Effective Pressure, in the Low-Pressure Cylinder, based on the Admission Pressure, and with the other Assumption given for Table E.

Actual Card No.	C (l. p.) Actual cut-off including clearance. Per cent.	H Average receiver pressure, gauge. Pounds per sq. in.	D 1 Theoretical mean effective pressure for l. p. cylinder, based on admission pressure, on 5 lbs. per sq. in. back pressure, and on the same compression that is found in Corliss engines. Pounds per sq. in.	O Actual mean effective pressure in l. p. cylinder. Pounds per sq. in.
1	87.3	58	52.3	50.9
2	84.2	54	49.5	46.2
3	78.5	54	49.5	44.8
4	71.2	50	47.4	32.5
5	61.6	47	42.8	24.4
6	61.6	46	43.8	20.1
7	55.1	46	39.0	16.4
8	55.1	45	37.2	13.2
9	55.1	45	39.8	15.4

15. Differences Between Actual Work done in Cylinder and the Work shown by Elementary Indicator Cards.—Table G shows the difference between the actual work done in both cylinders of the compound two-cylinder locomotives under consideration, and the work that would be given by calculation based on elementary indicator cards in which the steam was assumed to expand from the volume at cut-off in the h. p. cylinder, and with the pressure at admission in the h. p. cylinder, to the volume corresponding to the final volume of the l. p. cylinder, and illustrates the errors in some of the theoretical formulas offered for compound locomotives, more particularly in foreign technical publications. Such formulas as these have been used in argument about compound locomotives, and have generally led to conclusions entirely different from the results of actual trials of real locomotives.

To some extent this table also shows the loss in efficiency of compound locomotives due to inadequate valve motion, steam passages, and high speed, when compared to a good stationary compound engine, or a marine compound having better valve motion and running at slower speed.

TABLE G.

Showing the Difference between the Actual Work done in both Cylinders, and the Work that would be given by Calculation based on Elementary Indicator Cards.

Actual Card No.	Z Absolute pressure at admission, h. p. cylinder.	A 1 Absolute pressure at end of expansion, l. p. cylinder.	B 1 Actual work done in both cylinders per revolution, foot pounds, calculated for the purpose of comparing with column (C 1).	C 1 Theoretical work in both cylinders based on expansion in elementary engine from actual pressure at admission in h. p. cylinder, to pressure corresponding to final volume in l. p. cylinder, including clearance.
1	166	55	246,000	467,000
2	157	55	226,100	433,000
3	167	49	222,300	439,000
4	164	40	190,500	389,000
5	172	30	132,300	383,000
6	175	30	106,300	393,000
7	175	25	84,700	353,000
8	175	23	74,200	353,000
9	180	23	79,600	364,000

However, the difference in the power as given does not represent fairly the loss in efficiency. Loss in power does not necessarily indicate loss in efficiency; in fact, the loss in efficiency is very much less than the loss in power indicated by this table.

16. **Indicator Cards in Practice.**—In making a theoretical analysis of a proposed design of compound engine, the most important thing to do is to bear in mind the difference that exists between elementary indicator cards, on which such mathematical analysis is generally based, and actual indicator cards from practice. The causes which produce the differences are chiefly the initial condensation, re-evaporation during expansion, the size, shape and location of the steam passages and receiver; the opening of the exhaust before the end of the stroke; compression and wire-drawing due to the slow opening and closing of the ports, as well as the effect of the steam distribution by the existing types of valve motion. The following are some examples of the differences usually found between the elementary and the actual indicator cards:

17. Drop in Pressure During Admission, High-Pressure Cylinder.—Fig. 16 shows an indicator card from a compound locomotive in which steam was cut off at about $\frac{4}{10}$ of the stroke in both cylinders, as shown by the full line. The clearance space is 10 per cent. of the piston displacement in the h. p. cylinder, and 7.5 per cent. in the l. p.

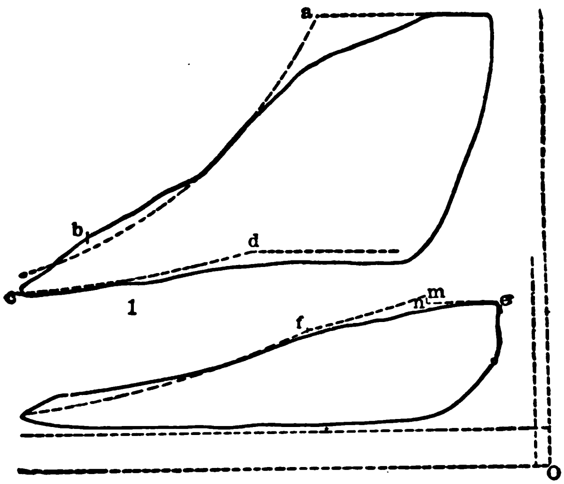


FIG. 16.
Cards Showing Drop of Pressure During Admission.

cylinder. The volume of the receiver is one and one-half times the h. p. cylinder. With this data the theoretical lines shown dotted in the figure have been constructed, making allowance for the excessive drop shown between the two cards. The differences between the actual admission and expansion lines of the h. p. card are the same as in cards from single expansion engines, and are due to the wire-drawing during admission and at cut-off, and to the re-evaporation during expansion.

18. Rise in Pressure During Admission, Low-Pressure Cylinder.—It will be seen from indicator cards, Figs. 14 and 15, that there is an increase in pressure in the l. p.

cylinder and in the receiver after the l. p. piston has moved somewhat from the end of the stroke. This is perhaps more pronounced in card No. 1, Fig. 14, taken at slow

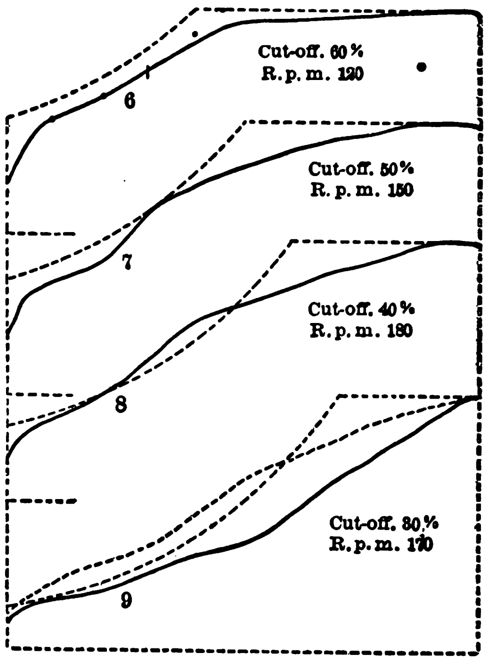


FIG. 17.

Difference Between Actual and Elementary Admission and Expansion Lines.

speed. This arises from the fact that the opposite end of the h. p. cylinder exhausts at this time, and thus increases the steam pressure in the receiver, and also in the l. p. cylinder. This action will always be found when the exhaust from the h. p. cylinder takes place before cut-off in the l. p. cylinder. This action is called "re-admission." It is not likely that with the ordinary valve gear, the h. p. exhaust in any compound locomotive will occur later than at 90 per cent. of the stroke, and the l. p. cut-off will not generally be earlier than $\frac{3}{10}$ of the stroke, and hence it is safe to say that re-admission will always occur in practice. The practical effect of this is to make the l. p. admission line more nearly parallel with the atmospheric line, or, in other words, causes the l. p. admission line to more nearly resemble the admission line of a card from a single expansion engine.

In Fig. 17 are shown the admission and expansion lines of four indicator cards from the l. p. cylinder of a compound locomotive. The points of cut-off given are those which were recorded on the cards. The dotted lines indicate the form of the theoretical card for these points of cut-off and for the initial pressures as shown.

On card No. 6 a curve which agrees with the actual curve very closely is indicated by dots, and shows an earlier cut-off than that recorded. On card No. 9 the irregular dotted line shows the form of the card from the other end of the cylinder with the same nominal point of cut-off.

19. **Effect of Speed on Shape of Indicator Cards.**—The extent of departures from the assumed theoretical curve varies greatly in simple engines, and principally depends upon the piston speed, valve gear, and size of steam passages. The only satisfactory way of determining the probable loss in a proposed engine, whether simple or compound, is to examine indicator cards from an existing engine of the same general proportions, and having a valve gear of the same type and dimensions. Indicator cards taken from engines of various makes when on similar service show variations of as much as 20 per cent., and it is obvious that no general rule can be laid down which will give the results that may be expected in any given case, as the conditions which affect the actual indicator cards are not only numerous but variable as well.

For example, in Fig. 16, when the h. p. exhaust occurs at b, the l. p. piston is at n, and re-admission to the l. p. cylinder takes place, causing a rise in pressure to m. The

l. p. piston moves from this position to that of cut-off *f*, $\frac{4}{10}$ of the stroke, before the h. p. piston has moved over the remainder of its stroke from *b* to *c*. The pressure at *c* was calculated approximately on the basis of the receiver pressure when the h. p. exhaust opened, being that at *f*. From *c* to *d* there is some compression as shown.

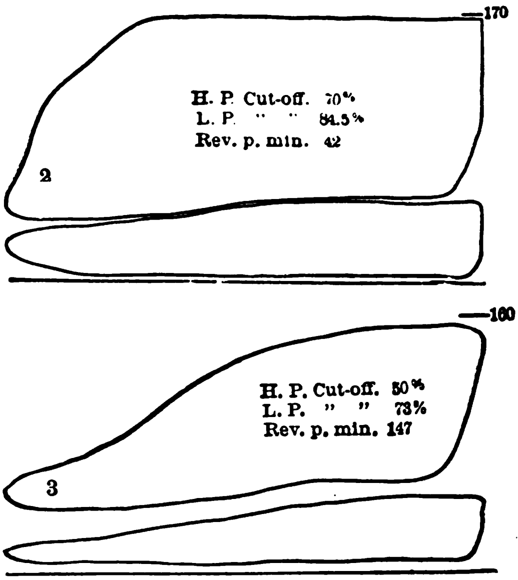


FIG. 18.
Actual Indicator Cards at Different Speeds.

Turning now to the l. p. card, and taking the pressure at *e* as that of the steam in the receiver, we find that the line from *e* to *n* is practically at constant pressure, and that the rise in pressure from *n* to *m* is comparatively slight. Also, that during the expansion of the steam in the receiver from *m* to *f* the fall in pressure is not great. The drop between the h. p. and the l. p. cards in this figure is excessive.

In Figs. 18 and 19 are shown indicator cards from two-cylinder compound locomotives at different speeds and points of cut-off. The shape of the h. p. back-pressure line is to be noted. Cards Nos. 2 and 3 are from the same engine, and it will be noticed that the compression up to about the middle of the back stroke is quite marked, and

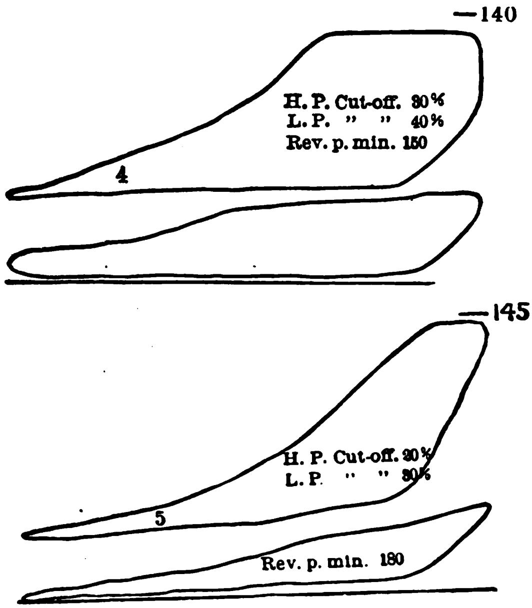


FIG. 19.
Actual Indicator Cards at Different Speeds.

that the remainder of the back pressure line is nearly horizontal, as it was found in Fig. 16. In Nos. 4 and 5 the compression appears to continue during the whole of the back stroke. This is the case in a considerable number of cards which have been examined, and is particularly noticeable at high speeds.

CHAPTER V.

EFFECT OF CHANGING THE POINT OF CUT-OFF—PRESSURE IN THE RECEIVER.

20. **Effect of Changing Cut-off in Elementary Engine.** —Perhaps the clearest way of indicating the general effect on the work done in the cylinders by changing the point of cut-off is to analyze the elementary engine and see the effect in it. In practice there is so much wire-drawing, particularly in the l. p. cylinder, that a change in the point

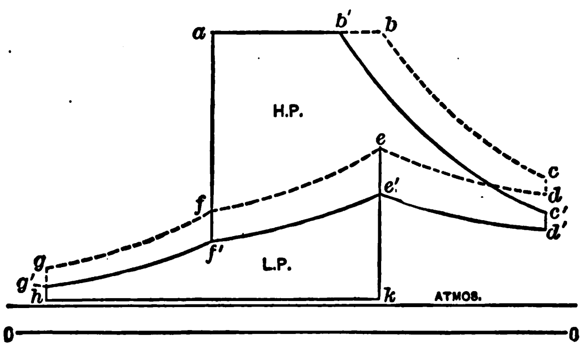


FIG. 20.
Effect of a Change in Point of Cut-Off.

of cut-off does not affect the power generated in the cylinder as much as in the elementary engine. Also, in cases where the receiver is small, a change in the cut-off in the l. p. cylinder is not always followed by a proportionate change in the mean effective pressure in that cylinder

To illustrate what takes place in the elementary engine when the cut-off is changed, reference is made to Fig. 1. Under the conditions assumed for that illustration, if the

h. p. cut-off is made earlier, while the l. p. cut-off remains as before, at one-half stroke, a series of changes will be introduced, which are shown in full lines in Fig. 20, the lines of Fig. 1 being repeated in dotted lines. Assuming a cut-off at $\frac{3}{8}$ stroke, the final pressure in the h. p. cylinder is $160 \times \frac{3}{8} = 60$ pounds, or at c' instead of c. Also, as the total expansion is now $2.5 \times \frac{8}{3} = \frac{20}{3} = 6\frac{2}{3}$ instead of 5, the final pressure at g is reduced to g', which represents $160 \times \frac{3}{20} = 24$ pounds. Then, as the l. p. cut-off is unchanged, the pressure at f is reduced to f', or $24 \times 2 = 48$ pounds. The steam which fills the h. p. cylinder at a pressure of 60 pounds is mixed with an equal volume in the receiver at a pressure of 48 pounds, giving a resulting pressure at d of 54 pounds. The results of this change are, then, that the pressure in the receiver, the initial pressure in the l. p. cylinder, and the mean pressure in that cylinder, are all less than before. The work done by the l. p. cylinder is therefore less, while for the h. p. cylinder we have taken from one part of the card and added to another part. The total work done by both cylinders is, of course, less than before, but the proportion done by the h. p. cylinder is greater, and, in fact, the mean effective pressure in that cylinder has been increased.

With both cut-offs at the same point, considerably more work is done in the l. p. than in the h. p. cylinder, but by making the h. p. cut-off the earlier of the two there is less difference in work than before, or, in other words, the work may be equalized by this means. A similar effect will, of course, be produced by making the l. p. cut-off *later* than that of the h. p., and conversely by making the l. p. cut-off earlier than that of the h. p. the proportion of the total work which is done by the l. p. cylinder will be increased. The following table, calculated for $R=2$ and $C=1.5\ v$, will illustrate this:

TABLE H.

Showing the Effect of a Change in Point of Cut-off in an Elementary Compound Engine.

Cut-off. h. p.	Cut-off. l. p.	Mean press. h. p.	Mean press. l. p.	Mean h. p. press. referred to l. p.	Total mean in one cyl.	Proportion of work. h. p.	Proportion of work. l. p.
$\frac{1}{2}$	$\frac{1}{2}$	46.6	54.0	23.3	77.3	.3	.7
$\frac{2}{5}$	$\frac{1}{2}$	51.4	39.6	25.7	65.4	.4	.6
$\frac{2}{5}$	$\frac{2}{5}$	39.2	48.9	19.6	68.4	.29	.71
$\frac{1}{2}$	$\frac{2}{5}$	31.5	60.3	15.7	76.0	.21	.79

21. Effect of a Change of cut-off on the Receiver Pressure in an Elementary Engine.— In locomotive practice the pressure in the receiver is less than that calculated, on account of losses in the h. p. cylinder and passages. The effect of a lower receiver pressure is to increase the proportion of work done in the h. p. cylinder, so that by adjusting the valve gear to give an earlier cut-off in the h. p. cylinder than in the l. p., the total work may be very nearly equally divided between the two cylinders of an elementary engine, and can be divided with sufficient approximation to equality in a well designed locomotive.

In Figs. 1, 2 and 3 the l. p. cut-off has been taken at one-half stroke, and it was assumed that release occurred in the h. p. cylinder exactly at the end of the stroke. If now we make the l. p. cut-off later than one-half stroke, leaving everything else unchanged, there will be an exhaust from the h. p. cylinder, while the l. p. steam valve is still open, which will increase the pressure in the receiver and cause what may be called a re-admission in the l. p. cylinder. This is illustrated by Fig. 22, in which the h. p. exhaust occurs at *b*, causing a rise in pressure to *c*, from which there is expansion as before in the h. p. cylinder, the receiver and the l. p. cylinder until the l. p. steam valve closes at *d*. A similar effect will be produced by pre-release in the h. p. cylinder. See Figs. 14 and 15.

An examination of a diagram such as Fig. 21 may make this subject more clear. In this Fig. *b c* represents the stroke of the pistons, and the circle the path of the crank pins. Taking the direction of revolution as indicated by

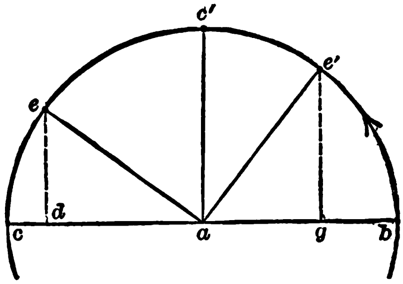


FIG. 21.
Diagram of Crank Location, Two-Cylinder Compound.

the arrow, when the h. p. piston is at the end of a stroke, or its crank is at *a c*, the l. p. crank will be at *a c'*, and the

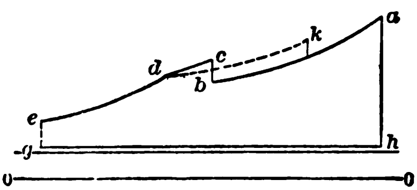


FIG. 22.
Rise in Pressure During Admission to l. p. Cylinder.

exhaust from the h. p. cylinder which takes place at this position of the cranks will cause the rise in the l. p. card shown at *c*, Fig. 22. If the h. p. exhaust occurs before the end of the stroke, for example when the piston is at *d*, the l. p. crank will be at *a e'*, and the l. p. piston at *g*, causing a rise in the l. p. card as shown at *k*, Fig. 22. In cards taken from an engine this increase in pressure will, of course, be more gradual, and at high speeds may simply cause the l. p. admission line to be more nearly parallel with the atmospheric line. This arises from the high piston speed and the consequent wire-drawing of the steam through the ports and past the valves.

22. **Equalization of Work in the High and Low Pressure Cylinders of a Receiver Compound.**—The nearer the action of the steam in a compound locomotive approaches the action in the elementary engine, the more readily can the power generated in the two cylinders be equalized at all cut-offs by an alteration of the cut-offs in the cylinders, and the reverse is also true; namely, that where the receiver is small and the wire-drawing and compression excessive, it is well nigh impossible to equalize the power generated in the two cylinders at all cut-offs by adjusting the cut-offs.

Some of the first compounds built in this country had much wire-drawing and compression, and had small receivers, and it was found practically impossible to equalize the power by changing the cut-offs. After some considerable experiment the receivers were increased and the compression was very considerably reduced by cutting out the inside of the steam valve, more particularly on the h. p. cylinder, so as to give what is termed "inside clearance" or negative lap, 80. On a 5½ inch travel, the amount cut out on each side was as much in one case as ½ of an inch. This clearance delays the point of exhaust closure and decreases the amount of compression. The result of these changes, when taken together with the longer steam ports now used, has been to put the two-cylinder compound locomotive at this time in very good shape, so far, at least, as the equalization of the work between the cylinders is concerned. This appears from Table I, for instance, which shows how perfectly the work is equalized in the Schenectady ten-wheel compound on the Central Pacific Railroad.

It is not expected that when a locomotive is starting a train and steam is used directly from the boiler in the l. p.

cylinders, that the work will be equalized in the h. p. and l. p. cylinders of any compound engine.

TABLE I.

Showing the Equality of Work in the High and Low-Pressure Cylinders of a Schenectady Two-Cylinder Compound Ten-Wheel Locomotive.

Cut-off h. p. Cylinder. Inches.	Cut-off l. p. Cylinder. Inches.	Per cent. of total work done in h. p. Cylinder.	Per cent. of total work done in l. p. Cylinder.
20¼	20⅝	45.0	55.0
19½	19⅞	45.8	54.2
17½	18⅜	46.5	53.5
15¼	16½	47.8	52.2
12½	14⅛	51.0	49.0
12½	14⅛	49.7	50.3
10¼	12⅜	48.5	51.5
10¼	12⅜	52.7	47.3
10¼	12⅜	48.5	51.5

23. Equalization of Work in the High and Low-Pressure Cylinders of a Non-Receiver Compound.—In the four-cylinder type of engine, which includes the tandem, Vauclain and Johnstone compounds, it is not necessary, either for the purpose of starting trains or for steadiness of motion of the engine, to equalize the work done in the cylinders. This appears from the fact that the two sides of the locomotive are duplicates of each other. However, in the Vauclain engine, in order to favor the peculiar construction of the crosshead, in which the centres of the piston connections do not coïncide with the centre of the main road bearing, it is very desirable to equalize the *pressure* at all parts of the stroke rather than the *work* done per stroke, and this brings in a new problem quite complicated in its nature, and which is not considered in the foregoing. This will be considered in the description of the Vauclain type of engine, as it has to do only with that particular construction, 121.

24. **Conclusions about the Equalization of Work in High and Low-Pressure Cylinders.**— In the two-cylinder receiver compound it is desirable to equalize the work done at all points of cut-off in the two cylinders except at starting, so that the difference will not be more than about 10 per cent. 20–23. In the tandem compound, it is not necessary or very desirable to equalize either the work in the cylinders or the pressures on the piston rod. In the Vauclain type of engine, 120, it is not necessary or very desirable to equalize the work done in the two cylinders, but it is quite necessary to approximately equalize the total pressures on the piston rods at different points of the stroke, in order to prevent a twisting tendency of the crosshead. This equalization cannot be made when steam is admitted directly from the boiler to the l. p. cylinder, yet it has been quite well equalized in some engines when running under normal conditions. In calculating the total pressures on the piston rod of the Vauclain engine to determine the equalization, it is necessary to include the pressures on the crosshead which result from the inertia of the piston, and this makes the calculations rather complicated. In a high speed engine, such as a locomotive, the inertia of the piston rod and piston modifies materially the total pressure on the piston rods. See Appendix P.

25. **Pressure in the Receiver.**—The variation of the pressure in the receiver, as shown on the lines *d*, *e*, *f*, Fig. 1, depends upon the capacity of the receiver compared with the capacity of the h. p. cylinder and the l. p. cylinder up to cut-off. For example, see Appendix E. As a further illustration of this, the following table shows the pressure at the points *d*, *e* and *f*, with receivers having capacity 1.5 and 2 times the capacity of the h. p. cylinder and with the l. p. cylinder capacity from 2 to 2.5 times the capacity of the h. p. cylinder, 53. It must be remembered that the results in this table are based upon

elementary indicator cards and not actual indicator cards, and are offered only in the way of illustration, and not for guidance in actual work, 12–19. The actual pressure in the receiver is materially modified by the action of the valve motion, the wire-drawing of the steam through the ports, and the compression in the l. p. cylinder.

	Pressure at *d*.	Mean press. bet. *d* and *e*.	Pressure at *e*.	Mean press. bet. *e* and *f*.	Pressure at *f*.	Mean press. in receiver.
$C = v$, $R = 2$.......	80.	91.8	106.7	91.8	80	91.8
$C = 1.5\ v$, $R = 2$.......	80.	88.9	100.	88.9	80	88.9
$C = 2\ v$, $R = 2$.......	80.	87.4	96.	87.4	80	87.4
$C = v$, $R = 2.5$......	72.	82.6	96.	77.8	64	80.2
$C = 2\ v$, $R = 2.5$......	69.3	75.7	83.2	72.4	64	74.

The table shows that the receiver pressure may vary during one stroke as much as 27 pounds, and that, generally, the pressure at *f*, the cut-off in the l. p. cylinder, will be below the admission pressure to that cylinder, and while it would appear from the table that the mean pressure up to cut-off, from *e* to *f*, does not differ much from the mean pressure in the receiver, yet, in fact, there is a considerable difference between these mean pressures, because of the wire-drawing of the steam through the port and past the valve of the l. p. cylinder. See Figs. 14 and 15.

In designing compound locomotives, the pressure in the receiver has been frequently assumed as constant. This assumption gives very simple formulas for receiver capacity and mean effective pressure, yet such formulas have no practical application, as the receiver pressure varies considerably in locomotive work owing to the irregular action of the valve motion and the wire-drawing and compression, 12 19. Some technical writers, more particularly in foreign publications, have deduced some quite simple mathematical expressions for the proper proportion of cylinder volume, receiver volumes, and points of cut-off, but these formulas have no practical application, for reasons that have been given, and because of further and incidental conditions that are imposed on locomotives. See Appendix K.

In most cases it is well-nigh impossible to pre-determine the receiver pressure by calculation, and the only safe way to proceed is to select actual indicator cards, of which there are now a great many available, from similar engines in practice, and make such changes in the actual cards as judgment and experience dictate, being guided in this by the differences between the proposed design and the actual similar design that has been tested in practical service. However, the table shows clearly one important fact. It is that the larger the receiver, the smaller are the variations of pressure in it. A further analysis of the practice in this respect is given under 45–56.

Upon the receiver pressure depends, to a great extent, the division of work between the cylinders, 50–51, and in an elementary engine or a slow moving locomotive the division of power may entirely depend upon this factor; but in an actual engine moving at considerable speed, the wire-drawing and compression so modifies the action of the steam that the control of the power distribution does not lie with the receiver pressure. Any useful rule for receiver pressures must necessarily be based almost entirely on the results from actual indicator cards, and will not be applicable to engines differing much in design.

If the pressure maintained in the receiver of an engine in practice is known, the probable receiver pressure in a similar proposed engine can be predicted; but when a quite different arrangement of valves and passages is used, the distribution in previous engines will be of little service as a guide in making estimates of receiver pressures.

When a compound locomotive is moving slowly, the wire-drawing and compression, 6–11, is not so much a factor in the distribution of power between the cylinders and in controlling the receiver pressure, and, therefore, an approximate calculation can be made with more satisfaction than for conditions when the locomotive is at speed. The following is a method of approximating to the probable receiver pressures at slow speeds:

$$p = c \times p_1 \frac{\text{h. p. cut-off.}}{\text{l. p. cut-off.}}$$

In this formula p is the absolute receiver pressure, p_1 the absolute h. p. initial pressure, and c is a numerical coefficient.

An examination of a considerable number of indicator cards from compound locomotives gave an average value for c of 0.46, but this value is not recommended except for approximations, and, of course, no such formula can take the place of direct experiment.

26. **Loss Due to Drop of Pressure in Receiver.**—The drop of pressure into the receiver, 25, represents an actual loss of efficiency, since it occurs by the expansion of the steam without doing useful work. For any given cut-off, or position of the reverse lever in a locomotive, this drop can be removed, but, in doing this, other losses or unsatisfactory actions at other cut-offs may result, which will make such removal of drop of receiver pressure at any particular cut-off undesirable. A method of calculating the drop in the receiver from elementary indicator cards, but which does not represent actual conditions, is given in Appendix E.

CHAPTER VI.

COMBINED INDICATOR CARDS AND WEIGHT OF STEAM USED PER STROKE.

27. **Combined Diagram Receiver Type.**—It is quite necessary, in order to understand where the losses are in compound locomotives, to construct what is called a "combined" indicator card, which is a diagram showing

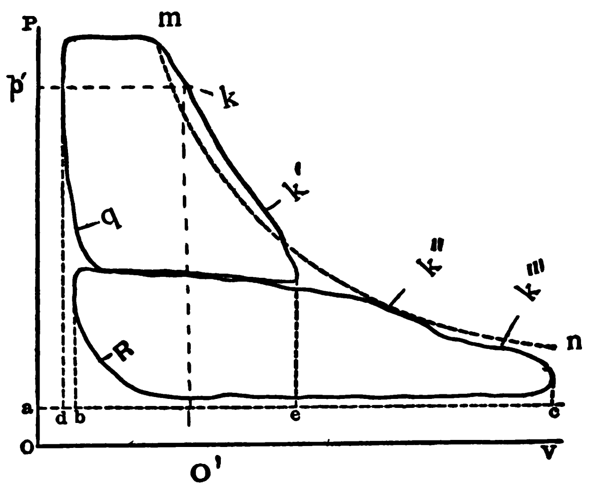


FIG. 23.
Combined Diagram from Two-Cylinder Receiver Compound.

the indicator cards from both h. p. and l. p. cylinders, drawn to the same scale and compared to a reference curve in the matter of expansion. In this way the expansion of the steam in the two cylinders is compared approximately with equal expansion in a single expansion engine, **45–46**; however, the usefulness of such diagrams is limited, and, at the best, they only show the serious defects, and not the minor ones.

28. **The Rectangular Hyperbola as a Reference Curve.**—The reference curve that is the most satisfactory of all to use is the rectangular hyperbola, 41, the method of drawing which has been described in Fig. 3. Fig. 23 illustrates a combined diagram from a two-cylinder receiver compound locomotive, of which the separate cards as taken closely resemble Fig. 19, card No. 4. In making this combined diagram, the cards are drawn to the same scale of pressures and volume as follows:

Take any convenient distance, such as *b c*, to represent the volume of the l. p. cylinder, and let *a b* represent the volume of its clearance space. Then *O a P* is the zero line from which to measure volumes, and *O V* drawn as usual is the zero line of pressures. Lay off *a d* equal to the h. p. clearance space, and *d e* equal to the volume of the h. p. cylinder, both on the same scale as that of the l. p. cylinder; or *d e* should equal *b c* divided by the ratio of the cylinders. The outlines of the cards are then found by plotting points as usual.

The rectangular hyperbola, *m n*, for instance, is not a curve that corresponds to equal steam weights at different points, but to the contrary, rises above the curve of equal steam weights, and therefore approximates more nearly to the real curve of expansion in the simple engine than the other curves of expansion sometimes used. See Fig. 23a. This explanation is necessary in order to indicate why the rectangular hyperbola is taken as the basis of such argument as is here offered about combined indicator diagrams.

It is evident that, at the point *K'*, the exhaust in the l. p. cylinder, all of the steam is not sent in the l. p. cylinder or receiver, but some of it is retained and is compressed in the clearance spaces; therefore, by calculating the amount of steam retained, say at q, we shall find a

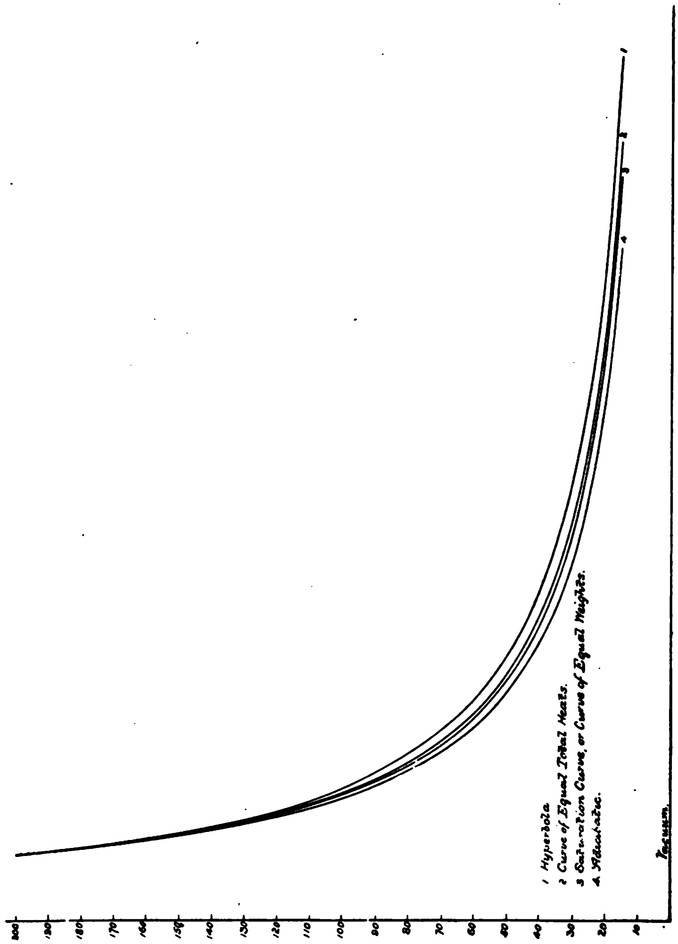


FIG. 23a.

Diagram Showing a Comparison of Hyperbola and Other Reference Curves.

substantial amount to be deducted from the amount at K', in order to get the weight of steam that is discharged into the receiver.

The actual weight of steam used per stroke is greater than the apparent weight, for the reason that the 15 to 40 per cent. of the entering steam that is condensed before cut-off is not re-evaporated during expansion, and the steam at K' contains a large amount of water, 69–72.

29. Location of Rectangular Hyperbola for Reference.—The point from which the hyperbola $m\ n$ should be

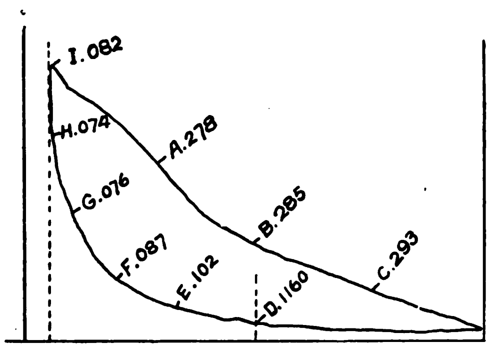


FIG. 23b.

Weight of Steam in Cylinder at Different Points of the Stroke.

drawn depends upon the purpose for which the examination is being conducted. Before further explanation of this, it is necessary to understand how much steam is used in a cylinder per stroke, and what should be expected of it in a comparatively perfect engine, 41–44.

30. Weight of Steam Used per Stroke.—By means of the total volume of the cylinder at any point, k, Fig. 23, which will be represented by $p'\ k$ and from the pressure of the steam represented by $o'\ k$, the total weight of the steam in the cylinder at k can be calculated. This is true of other points, $k'\ k''$ and k''', also of q and R. In a single expansion engine it will be found, by calculation from an actual indicator card, that the weight of steam increases from k to k''' almost uniformly, see Fig. 23b, 42–44. This is due to the re-evaporation during expansion of the steam that was condensed before cut-off, due to the cooling effect of the cylinder walls. The re-evaporation is caused by the heating effect of the cylinder walls on the steam and water in the cylinder. As the pressure falls during expansion, the temperature of the steam falls, and the walls, being hotter than the steam, re-evaporate some of the moisture in the cylinder, 69–72.

We have seen that the steam sent to the l. p. cylinder from the h. p. is the difference between that at k' and q. If none of this steam is lost in transit through the receiver or in entering the l. p. cylinder, it will be apparent in that cylinder, and the difference between the steam at k'' and the steam at R should equal that sent from the h. p. cylinder. Later on, at k''', it should be expected that further re-evaporation would make more steam apparent. This can be learned from the difference between that at k''' and R than that between k'' and R. This continued re-evaporation in the l. p. cylinder generally takes place, and in a good compound locomotive, where the valves are tight, it will be found that the steam present, as shown by the indicator cards, will increase quite regularly from the point k to the point k''', when allowance is made for the steam retained in the h. p. cylinder at q, 44

31. **Weight of Steam Retained in Cylinder at End of Compression.**—In assuming or locating the points q and R, much care should be taken, as the amount of steam in the cylinders, shown by the indicator cards, decreases continually from the time the exhaust closes, which is the commencement of compression, to the opening of the valve for pre-admission due to lead, 6. The point q should be taken to represent, as nearly as possible, the weight of steam in the cylinder when the valve opens, and, there-

fore, it should be taken well up on the compression line, and as near to the point of admission as possible. This is also true of the point *R*. Fig. 23b further illustrates this, and shows the change in apparent steam weight during compression. See Fig. 4.

It is clear that if the valves of a compound engine are tight, the same amount of water, in the shape of moisture, steam and water, must be discharged from the h. p. as from the l. p. cylinder at each stroke ; otherwise, if the l. p. discharged more than the h. p. the receiver would be quickly emptied, or if less than the h. p. it would be quickly filled with water and steam, 44. All this adjusts itself automatically, and the pressure in the receiver rises and falls as the cut-offs in the cylinders are changed in such a way as to bring about the same discharge of water, in the shape of steam and moisture, from the l. p. cylinder as is discharged from the h. p. cylinder into the receiver.

32. **Limitations of Combined Diagrams.**—In making an examination of the action of an engine, by means of the combined diagram, it must not be forgotten that such diagrams have a distinct limitation, which is found in the fact that they show only the steam in the cylinder and, therefore, only the apparent amount of water, and do not show the moisture or water in the cylinder, which must be added to the apparent amount of water, in the shape of steam, in order to get the actual total water used per stroke, 69–72. In other words, there is a considerable amount of water passing through the cylinders of the compound engine, in the shape of moisture in the steam, which is not measured, indicated or made apparent by the indicator cards, 69. However, this limitation of the value of combined diagrams does not prevent them from being decidedly useful when such limitation is understood and allowed for, as will appear from what follows:

33. Re-evaporation in Receiver.—If in a compound receiver engine it is found by calculation from the indicator cards, 30, 72, that more apparent water, in the shape of steam, is used per stroke in the l. p. cylinder than in the h. p., then one may be led to understand that there is either a leakage in the valves or a re-evaporation (not super-heating) in the receiver.

Super-heating in the receiver of a compound locomotive is practically impossible, unless the smoke box temperature is above what it should be for good economy in the boiler, for the reason that the steam passes through the receiver when the engine is at speed at a rate that would make it impossible to collect enough heat to re-evaporate all of the moisture in the steam, much less to cause a super-heat, 54–55. This has been shown by tests made by Mr. William Forsyth, Mechanical Engineer, of the Chicago, Burlington and Quincy Railroad, on a two-cylinder compound locomotive having a receiver in the smoke box. It is true that the temperature of the smoke box is about 600 degrees Fahrenheit, quite sufficient to produce a substantial super-heat, if the steam remained in the receiver long enough to permit it; but at 200 revolutions per minute, which is an ordinary velocity for a locomotive, there are 400 exhausts into the receiver per minute. If the receiver is about twice the volume of the l. p. cylinder up to cut-off, then each cubic foot of steam remains in the receiver about $\frac{1}{200}$ part of a minute, or about $\frac{1}{3}$ of a second, a much too short time to permit of super-heat.

34. Condensation in Receiver.—On the other hand, if it is found that less steam is apparently used in the l. p. cylinder than is discharged into it from the h. p. cylinder per stroke, then it may be expected that there is a loss of steam by condensation in the receiver or in the l. p. cylinder, 54–55. Some results of calculation of this kind are given in Table J. 30, 72.

In this way an examination can be made to learn if the steam at k'', Fig. 23, less that at R, is greater than that at k', less that at q. This will indicate whether there is a gain or loss up to cut-off in the l. p. cylinder. Allowance should, of course, always be made for the steam at q and R, as the steam at R always mixes with the incoming steam from the h. p. cylinder. To be still more accurate, the difference in the heat contained per pound of the steam at R, q, k'', and k', should be allowed for.

35. **What is Shown by Reference Curve on Combined Diagrams.**—It now will be clear that in drawing the rectangular hyperbola $m\ n$, it may be drawn from the point k to note the re-evaporation at k', or from some point, as m, located so that the volume $P\ m$ corresponds to the volume of the weight of the steam, which is discharged into the l. p. cylinder at each stroke. Manifestly, when the curve $m\ n$ is located in this way, it will fall to the left of k', Fig. 23, and if there is no loss between the cylinders and up to cut-off in the l. p. cylinder, it will pass just to the left of point k'' and inside of the expansion curve of the l. p. cylinder by an amount which depends upon the steam that is added to the incoming steam from the h. p. cylinder, from the compression or clearance spaces in the l. p. cylinder. This last amount is that which is calculated for the point R. This is further explained in the analysis of the combined diagrams from the four-cylinder non-receiver type, 41–44.

36. **Ideal Combined Diagram.**—To show what the ideal combined indicator card would be from a compound, reference is made to Fig. 24. This card was taken from a triple expansion Corliss pumping engine running at twenty revolutions per minute. The cylinders were 5 feet stroke, and with the following diameters: H. p. cylinder, 28 inches; intermediate cylinder, 48 inches; l. p. cylinder, 74 inches. Careful tests of this engine

showed a consumption of twelve pounds water per horse-power per hour. There is little, if any, loss of steam by the drop in the receiver, and practically no loss from compression and wire-drawing. Compound locomotives cannot be made to give cards like this, even at the slowest speed, for the reason that the locomotive has to be designed to work at different cut-offs, while the stationary compound is made principally for a single cut-off, or with very small variations therefrom. However, a comparison of this card with an actual indicator card, Fig. 25, will show where the loss occurs in the compound locomotive at the

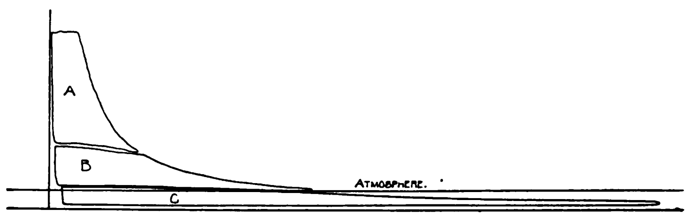


FIG. 24.
Ideal Combined Card.

present time, and further explains why high speed compound locomotives have not given the economy that they should, 139–147.

The upper cards *A* and *B* of this diagram, Fig. 24, represent probably the best steam distribution that has been obtained from a two-cylinder receiver compound. Taking the area of these cards *A* and *B* and calculating the horse power, omitting the l. p. card *C*, and taking the same total water per hour that was actually used in the test, the water per horse power is found to be 18 pounds. That is to say, while the water per horse power per hour with the triple expansion engine, giving cards *A*, *B* and *C*, is 12 pounds, yet by omitting the work done by card *C*, to bring the result more nearly like a two-cylinder compound locomotive, the resulting water per horse power per hour is about 18 pounds. It may be said then that a compound locomotive must use steam with approximately as good distribution as shown by Fig. 24, in order to reach as low a water rate as 18 pounds per horse power per hour. However, the steam pressure on a locomotive is generally higher, say 180 pounds per square inch, while in the case of Fig. 24 the steam pressure was but 120 pounds. On the other hand, the triple expansion engine had steam jackets and other advantages which would tend to offset the advantage of higher boiler pressure.

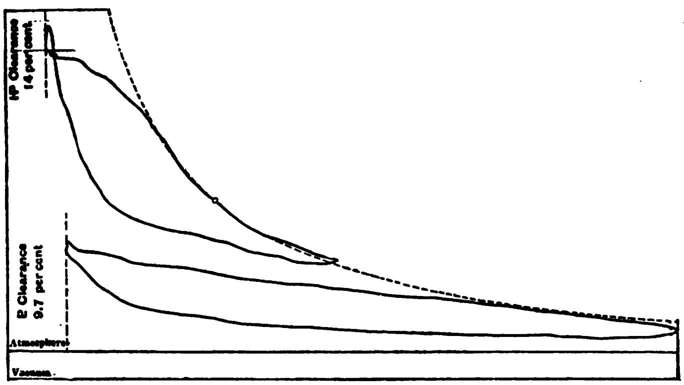


FIG. 25.
Actual Combined Card.

37. Combined Diagram from Non-Receiver or Woolf Type.—Combined diagrams from the Woolf type of compound having no receiver, sometimes called "continuous expansion" compounds, differ greatly in appearance from those of receiver compounds, 27–32, as will appear from Figs. 23 and 26. The following is an analysis of Fig. 26, which will emphasize what has been said about steam use for Fig. 23. The cards in Fig. 26 have been combined on a new plan, which shows the effect of clearance in the cylinders and valves. The line *ZC''* is the line of zero pressure. The line of atmospheric pressure is just above it. The mean effective pressures and the clearances of the engine are given on the diagram. The indicator cards, shown on the left hand part of the diagram, are an exact reproduction of the ones taken from the engine. The indicator diagram on the right side shows the l. p. diagram enlarged, so that the pressure at each individual point of the diagram is plotted on a volume exactly equal to the volume which the steam occupied in the l. p. cylinder when it had a corresponding pressure. For instance, take the point K on the l. p. diagram, the pressure represented by $G' K$ is exactly that which was in the l. p. cylinder at admission, and is equal to $F' Y$, while the volume which is represented by the distance $O G'$, is exactly the volume which the steam occupied in the cylinders when it has the pressure, $G' K$, and this is true of every other point on the expansion line of the combined diagram.

38. **Method of Combining Indicator Cards from Non-Receiver Type.**—The method of combining the diagrams is as follows :

From O, which is the point of zero volume, the distance $O C'$ is laid off equal to the h. p. clearance. $C' F'$ is the length of the indicator card as taken. $F' P'$ corresponds to the volume of the space in the valve between the h. p. and the l. p. cylinders. $P' G'$ corresponds to the clearance in the l. p. cylinder. OC'' corresponds to the volume of the l. p. cylinder (being about 2.93 times the volume of the h. p. cylinder), plus the l. p. clearance. Between the vertical lines drawn from $C' F'$ the actual indicator card is laid out.

The line $K H$ is the expansion line in the l. p. cylinder taken from the actual indicator card, and the pressure at every point on this expansion line is plotted at a volume point exactly corresponding to the volume of the steam in the cylinders, as shown by the actual indicator cards. At the point *H*, which is the cut-off in the l. p. cylinder, the volume is reduced by the amount *H J*, which is the sum of the volume of the interior of the valve, or *R Q*, and the volume remaining in the h. p. cylinder and the volume of h. p. cylinder clearance together, or *V U*. Thus the volume occupied by the steam after cut-off is represented by the distance *O M'*, and the pressure corresponding to that volume is *M' J*.

After cut-off the steam expands from the point *J*, as shown by the line *J I'*, and this line corresponds with the expansion line on the actual indicator card; that is, at each point the pressure is plotted on a volume corresponding to the actual volume occupied by the steam.

This method of plotting is necessary in order that a comparison may be made between the lines *EE'-EE''-DD'* and *D D''*, which are theoretical lines drawn to show any peculiarities of the expansion of the steam in the two cylinders, 43. Without this method of plotting no fair comparison could be made, as the pressure would not be plotted on actual volumes, and a false and untrue condition would be exhibited.

The over-lapping of the l. p. indicator card from *H* to *J* is necessary by reason of the abrupt reduction in the volume occupied by the steam at cut-off in the l. p. cylinder, the reduction being caused by the cutting out of the volume of the valve and the volume yet remaining before the completion of the stroke of the h. p. cylinder. In order that the true area of the combined indicator card may be preserved, it has been found convenient to draw the dotted sections *R Q P S* and *V U T W*, which are together equivalent to *J I N M*. This makes the mean effective pressure determined from the entire area of the combined indicator cards, including the dotted section, exactly the same as that determined from the original cards.

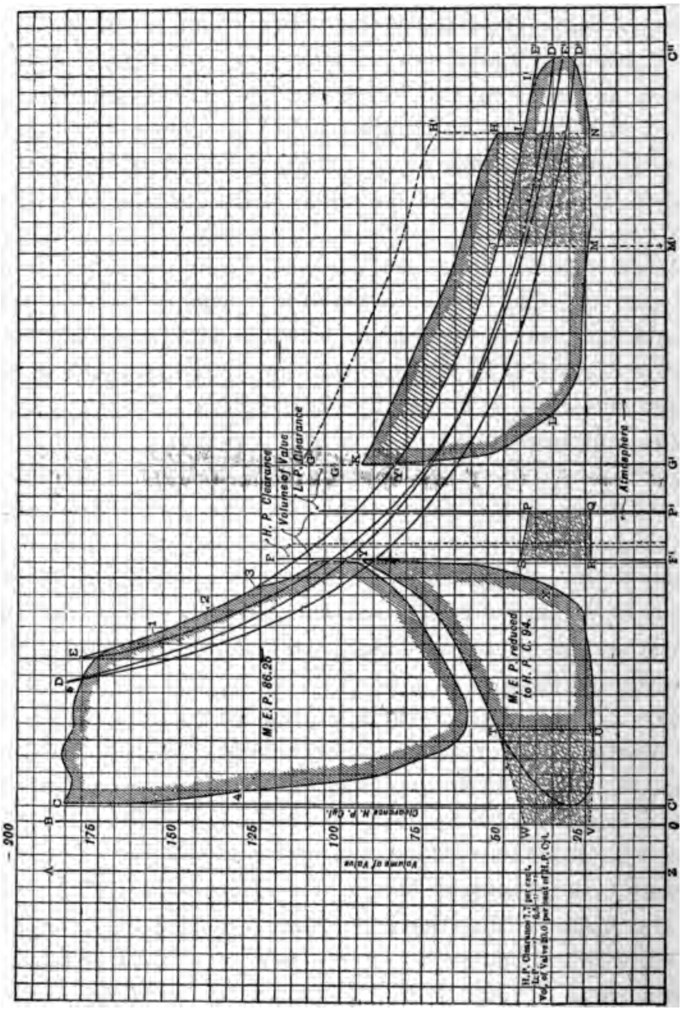


FIG. 26.
Combined Diagram from Non-Receiver or Woolf Type.

The pressure during exhaust and compression on the combined diagram is plotted at the same point as the corresponding pressure in the steam line of the actual card; that is to say, the back pressure at *N* is the one corresponding to the back pressure on the point below the cut-off point on the original indicator card. That is, the pressure at *N* is the same as the pressure at *U*, just as the pressure at *H* is the same as the pressure at *T*. This is an unimportant fact, however, as the combined diagram is mainly drawn for the purpose of examining the correspondence between the theoretical expansion line and the actual expansion line of the steam in the cylinders, and not to get the mean effective pressures. By these lines are shown the continual re-evaporation and corresponding increase in apparent steam weight during expansion in the h. p. cylinder. At the point 3 the h. p. cylinder exhausts into the valve and into the l. p. cylinder clearance. Here it meets with steam that was retained in the valve at cut-off at the point *H* or *T* in the l. p. cylinder, and with steam that was left in l. p. cylinder clearance after compression, and therefore the total steam weight is increased.

If no steam leaked out of the valve or condensed from the time it was shut in at cut-off in the l. p. cylinder, and none of the steam was condensed or lost from the clearance spaces after compression in the l. p. cylinder, the total steam weight at the point *K* would be the sum of the steam exhausted from the h. p. cylinder, the steam that was left in the valve, and the steam remaining in the l. p. clearance.

39. **Losses Shown by Combined Diagram from Non-Receiver Type.**—If there were no losses, and making due allowance for the lower pressure and temperature of the steam in the valve and in the l. p. clearance, the pressure at *K*, Fig. 26, should be 101 pounds absolute instead of 92 pounds. The weight of the steam in the valve and in the l. p. clearance, which would be mixed with the steam

from the h. p. cylinder, at exhaust from the h. p. cylinder is about 21½ per cent. of the weight exhausted from the h. p. cylinder, provided there was no loss of any kind from the clearance of the l. p. cylinder and the clearance in the valve after the steam was shut into these cavities. The point *G* shows what the pressure would be if there was no loss. If all the steam shut in was lost, then the point *K* would fall about to the point Y′. The tighter the valve and the less the loss in other ways of the steam that is shut in, the higher the point *K* will be above the point Y′. It has been said that the rise of pressure at the point *K* above Y′ shows leakage, but this is a mistake, unless all the steam shut into the valve and into the l. p. clearance is assumed to be lost. That this steam is not wholly lost is shown by the fact that the point *K* does actually rise considerably above the point Y′.

As we go on with this analysis to the point of cut-off, or at *H*, we find that the weight of steam in the cylinders, as shown by the indicator card, increases continuously and according to the following numbers :

Weight at *K*, .66 pounds ; and at other points, .66, .68 and at the point *H* .70 pounds. At this point the volume is decreased by *H J*, and steam at the pressure *H* is shut into the valve and the h. p. cylinder, and the total apparent steam weight is decreased, as shown by the actual indicator card, to .575 pounds, the pressure, of course, remaining the same as at *H*.

In the case of this particular indicator card, it is curious to note that the point *J* falls upon the hyperbola *E Y′* E′ drawn from the h. p. indicator card expansion line, and indicates that, up to the point of cut-off in the l. p. cylinder, there has not been leakage enough or re-evaporation enough to raise the steam pressure above the hyperbola drawn from the expansion line of the h. p. indicator card.

Also it is a curious fact that in this particular indicator card the expansion line in the l p. cylinder after cut-off, as shown by *J I*, corresponds almost exactly with the hyperbola *E E'*, just described. This shows that while at the point of the exhaust from the h. p. cylinder a considerable amount of steam is added to that exhaust (from the interior of the valve and from the l. p. clearance), yet this added steam is not wholly lost, but part is returned again to the valve and h. p. cylinder at the point of cut-off in the l. p. cylinder.

As has been said before, 28, the hyperbola corresponds more nearly to the actual expansion line of steam in a locomotive cylinder than does the adiabatic, owing to the re-evaporation of the steam that was condensed up to the point of cut-off. Therefore, if the point *J* on any combined indicator card should fail much below the hyperbola *E E'*, one would suspect considerable loss due to condensation; and if it should rise very much above this hyperbola, one would suspect leakage or an unusual amount of re-evaporation, but more probably leakage.

40. **Correct Area of Combined Diagram Non-Receiver Type.**—In measuring the area of this combined indicator card, one must follow the lines *K H I E" N M L K*. This will appear from a study of the way in which the card is laid out. This method of combining cards is exceedingly simple and can be followed without inconvenience. To do it one needs only to calculate the volume occupied by the steam at several points and plot these volumes from *O* as an origin.

41. **Reference Curve for Combined Diagram Non-Receiver Type.**—The proper theoretical line to be drawn for comparison on a combined indicator card is a matter of some dispute, but as each line has its own particular value and meaning, there is not much to dispute about, 43. The *point* from which the theoretical line should be drawn is of more importance.

In a single expansion engine with tight valves, the total amount of water in the shape of steam and moisture in the cylinder does not change after cut-off until exhaust is reached. Some of the steam may be condensed, but the total water remains the same. With compound engines, of the non-receiver type, however, this is not so, for the reason that at cut-off in the h. p. cylinder and at the closure of the exhaust from the h. p. cylinder, a considerable amount of steam is retained in the valve and clearance of the h. p. cylinder. The steam used per stroke in the h. p. cylinder, as apparent from the indicator card, is the difference between the amount present in the cylinder at the point 3, Fig. 26; and the amount retained in the cylinder during compression, taken for example at the point 4. For one to draw the theoretical steam line from the point 3 is to assume that all the steam that enters the h. p. cylinder during admission is exhausted therefrom, but this is not true. The real amount is the difference just referred to, and is represented by the volume *B D*, *B E* being the amount admitted to the h. p. cylinder; so that to look for leakage or re-evaporation in the l. p. cylinder after cut-off, the theoretical steam line should be drawn from the point *D*, and not from the point *E*.

42. **Weight of Steam per Stroke.**—It may not be clear why this is so without further explanation. In any compound engine as much water in the shape of steam or moisture must pass out of the l. p. cylinder as is passed out of the h. p. cylinder; otherwise, there will be a collection of water in the l. p. cylinder which would go on until the cylinders were full. That is, the amount of water in the shape of steam taken from the boiler at each stroke of the h. p. cylinder must be the same as that thrown out from the l. p. cylinder at each stroke.

If the volume *B D* and pressure at *D* indicates the amount of steam given from the h. p. cylinder to the l. p. at each stroke, then this amount should be looked for after the cut-off in the l. p. cylinder, barring, of course, all gains due to re-evaporation of the moisture in the steam and the losses due to any condensation, 69, that may take place. This leads to the conclusion that in an examination of the steam lines on the combined card from *E* to *H* (*H* being the point of cut-off in the l. p. cylinder, and also the point of the commencement of compression in the h. p. cylinder), the theoretical expansion line should be drawn from the point *E* and for the examination of the steam pressures after cut-off in the l. p. cylinder, that is, from *J* to the end of the stroke, the theoretical steam line should be drawn from the point *D*. It follows, then, that to determine, by comparison of pressures at the end of the expansion of steam in the two cylinders, the leakage, re-evaporation, or condensation, during the passage of the steam through the cylinders, the theoretical steam line should be drawn from the point *D* and the comparisons should be made after cut-off in thel. p. cylinders. This is because any leakage, re-evaporation, or condensation, will show up most prominently after the cut-off point *J*, Fig. 26.

43. Other Reference Curves for Combined Diagrams.—In this particular diagram both the hyperbola and the adiabatic lines have been drawn from both points *E* and *D*. $E\,E'$ and $D\,D'$ are hyperbolas. $E\,E''$ and $D\,D''$ are adiabatic curves. It will be seen that the point *J* rises considerably above the adiabatic curve drawn from *D*, and this shows either some leakage or re-evaporation. It also falls somewhat above the hyperbola from the point *D*. This is a further indication of leakage or re-evaporation; but there is and should be in every engine a considerable amount of re-evaporation, which will frequently raise the actual steam line above the hyperbola. Therefore, so far as this combined diagram shows, there is no strong evidence of leakage. However, the combined diagram is not the best way to show leakage. It is a good graphical way of showing how the volume, pressure and weight of steam changes during the entire expansion of the steam, but it is not as accurate in showing leakage or re-evaporation as the comparison of the steam weights. See Appendix O.

44. Weight of Steam per Stroke, Various Compound Locomotives.—Take this particular card and refer to Table J, Card No. 3, C. B. & Q. tests. It will be seen that the card shows that .507 pounds of steam was used per stroke in the h. p. cylinder and .493 pounds used per stroke in the l. p. cylinder. These amounts are practically the same, and, so far as the indicator card goes, there is no evidence of more steam being thrown out of the l. p.

TABLE J.

Giving the Weight of Steam Used per Stroke in Several Compound Locomotives. This Data was Calculated from Sample Indicator Cards.

Engine.	Indicator Card.	Speed in Rev. per Minute.	Weight of steam used per stroke, h. p. cyl.	Weight of steam used per stroke, l. p. cyl.	Increase of weight of steam in l. p. cyl.	Decrease of weight of steam in l. p. cyl.	Per cent. of increase weight of steam in l. p. cyl.	Per cent. of decrease weight of steam in l. p. cyl.	Cut-off in h. p. cyl., per cent. approximate.
Baldwin No. 82 in C., B. & Q. tests.	51	121.4	.4999	.5026	.0027	......	0.5	....	67
	48	140.1	.5000	.4931	....	.0069	....	1.4	63
	8	210.2	.3428	.3650	.0222	......	6.5	....	58
	3	140.1	.5071	.4908	....	.0163	....	3.2	64
	33	186.8	.3780	.4052	.0272	......	7.2	....	58
Baldwin No. 82 in Erie tests.	1	120	.4568	.4451	....	.0117	....	2.5	58
	2	160	.4320	.4452	.0132	......	3.0	....	57
	3	160	.4293	.4007	....	.0286	....	6.6	56
	4	140	.3499	.3564	.0065	......	1.9	....	48
	5	172	.3605	.3681	.0076	......	2.1	....	52
Schenectady, 12 Wheeler, Eng. No. 367.	2&2a	150	.8022	.7103	....	.0919	....	11.46	44
	6&6a	156	1.1374	1.0646	....	.0928	....	8.02	66
	7&7a	180	1.0347	.9994	....	.0453	....	4.38	55
	8&8a	192	.9518	.8579	....	.0939	....	9.86	51
Schenectady, 10 Wheeler, Mich. Cent.	61	100	.7920	.7144	....	.0776	....	9.80	59
	74	152	.7028	.6363	....	.0665	....	9.46	59
	80	124	.8247	.7259	....	.0988	....	11.98	59
C., B. & Q. Mogul, Eng. No. 324.	8	243.9	.5861	.6631	.0770	......	13.1	....	41
	49	173.4	.4932	.6001	.1069	......	21.6	....	35
Rhode Island Comp. on Brooklyn Elevated.	27	180	.2765	.2639	....	.0126	....	4.0	86
	12	306	.1576	.1421	....	.0155	....	9.0	64
Great Eastern Worsdell Comp., Eng. No. 230.	3	252	.5110	.4586	....	.0524	....	10.	42
	4	192	.4992	.4609	....	.0383	....	7.7	57
	5	264	.3918	.3390	....	.0528	....	13.5	48
Mexican Central, Johnstone Comp.	1	72	.5930	.6918	.0988	......	16.6	....	69
	2	60	.5947	.6611	.0664	.....	11.2	....	70
	3	57	.6344	.7140	.0796	......	12.5	....	79
	4	66	.5887	.6399	.0512	......	8.7	....	70

cylinder than is thrown out of the h. p. cylinder, which would be the case if there was any considerable leakage through the piston valve. In making these analyses one must remember that there is a large amount, something over 30 per cent., of water present in the steam at the point of cut-off in the h. p. cylinder, and the major part of this water goes through the engines without being shown on the indicator card. It is this water which re-evaporates and raises the steam line at cut-off in the l. p. cylinder above the adiabatic curve. We have seen that in this card there is no more steam used by the l. p. cylinder than by the h. p., but this is also true of other cards from this and other engines of the same type, as shown by Table J. As the pressure of the steam decreases during expansion there is a continual increase in apparent weight from the indicator cards.

If the rate of re-evaporation in the h. p. cylinder (if such it be and not leakage, and it probably is re-evaporation, as there is no reason to believe that steam would not re-evaporate in this type of h. p. cylinder just as in any other h. p. cylinder) be continued until the commencement of the stroke of the l. p. cylinder, the weight of steam at *K*, Fig. 26, would correspond to the actual apparent weight from the indicator card. But it is not to be expected that this rate of re-evaporation would thus continue, owing to the fact that the steam when it is discharged from the h. p. cylinder meets comparatively cold surfaces and intermingles with steam in the valve and in the l. p. clearance which is of a lower temperature. Of course this last argument is mainly a speculation, and is interesting only so far as speculation goes. It is a curious fact, however, that assuming the rate of re-evaporation to continue, the calculated weight of the steam shut into the valve and l. p. clearance would raise the pressure to G'' at the commencement of the stroke of the l. p. cylinder, and the loss would have been $G'' K$, but that it is impossible that this was the case is clearly seen from an analysis of the steam weight at different points of the indicator card. To claim that the valve, at the time of admission to the l. p. cylinder, is filled to the same pressure as the pressure of the exhaust from the h. p. cylinder, as has been claimed, is to admit that the area represented by $G'' K H H'$ is wholly lost. But it is easily shown that this is not the case.

The indicator card, Fig. 26, shows that about 17.4 pounds of steam were used per horse-power per hour. Of course this does not account for the loss due to condensation up to cut-off. From the actual tests an approximate estimate of the water used per horse-power per hour is 29.9 pounds, leaving 10.5 pounds of water per horse-power per hour not shown by the indicator card, the measurements being taken just after cut-off in the h. p. cylinder. This indicates a condensation of about 37 per cent. of the steam entering the h. p. cylinder up to cut-off. The insufficient data from which this result is obtained renders it probable that the 37 per cent. is not the correct amount. It may be more, but it is probably less. This, of course, is only another speculation and interesting only so far as speculations go. However, the plan of analysis indicates what can be done when a complete set of data is furnished. Whether this data can be collected from a road test is somewhat uncertain, but it surely can be collected from a shop test, such as is now made regularly at the Purdue University by Professor Goss, who has a large Schenectady single-expansion eight-wheel locomotive mounted on carrying wheels and operated with as much power as the same engine would exert if it were hauling a regular train. The advantage of this arrangement is that very accurate measurements can be made of the water and fuel used. It also permits accurate indicator cards to be taken.

CHAPTER VII.

TOTAL EXPANSION. RATIO OF CYLINDERS.

45. Total Expansion from Elementary Indicator Cards.—It is frequently necessary, for the purpose of comparing the action of locomotives, to know the total expansion of the steam in each type, and while it might appear from Figs. 1 and 2, 2–3, that this can be done by reasoning from the known volumes of the cylinders and points of cut-off, yet in fact the steam use is so affected by wire-drawing and compression that calculation is of little or no value, 12–19. The only accurate way to get the total expansion is to examine the actual indicator cards, from a locomotive that has been built, or the pre-determined indicator cards of a proposed design. These pre-determined cards should always be made up from cards from existing engines of similar design, with such corrections as experience or judgment show to be necessary to include the differences between the proposed and actual locomotives. An approximate method of calculating the total expansion from the elementary indicator card is given in Appendix D.

46. Total Expansion from Actual Indicator Cards.—The difference which is generally found between the theoretical* total expansion and the actual total expansion in practice is shown by Table K.

Table K, taken from same data as Tables B, C, D, E, F, and G, shows the difference in the ratios of expansion in the individual cylinders and the total in both cylinders when estimated by different rules commonly used, and illustrates

* "Theoretical" as here used is intended to be understood as applying to the limited theory of steam expansion commonly used as a basis for the computation of mean effective pressures. See Chapter I.

TABLE K.

Showing the Difference in the Ratios of Expansion in the Individual Cylinders, and the Total in both Cylinders when Estimated by Different Rules Commonly Used.

Card Number.	C Actual cut-off including clearance, per cent. H. P.	C L. P.	D Apparent cut-off, per cent. H. P.	D L. P.	Q Ratio of expansion h. p. cylinder based on actual cut-off.	R Ratio of expansion l. p. cylinder based on actual cut-off.	S Total ratio of expansion both cylinders based on actual cut-off.	T Actual total ratio of expansion measured from indicator cards. Being ratio of terminal to initial pressures.	U Per cent. of increase of actual total expansion above the total expansion based on actual cut-of in both cylinders.	V Total ratio of expansion both cylinders based on actual cut-off in h. p. cylinder and ratio of cylinders. Being product of Col. Q. and ratio of cylinders which is 21.	W Ratio of expansion in h. p. cylinder based on apparent cut-off.	X Total ratio of expansion both cylinders based on apparent cut-off in h. p. cylinder and ratio of cylinders, showing the error of this basis and the greater apparent expansion when apparent, and not actual cut-off is considered.	Y Pressure at end of expansion in l. p. cylinder. Pounds per sq. in.	Z Absolute pressure at admission h. p. cylinder.	A1 Absolute pressure at end of expansion l. p. cylinder.
1	86.4	87.3	84.5	86.0	1.16	1.15	1.33	3.0	204	2.44	1.18	2.48	40	166	55
2	83.8	84.2	81.2	82.8	1.19	1.19	1.42	2.85	201	2.5	1.23	2.57	40	157	55
3	76.6	78.5	73.0	76.8	1.31	1.27	1.66	3.4	205	2.75	1.37	2.88	34	167	49
4	68.2	71.2	63.5	68.8	1.47	1.40	2.06	4.1	199	3.2	1.57	3.30	25	164	40
5	58.4	61.6	52.1	58.5	1.71	1.62	2.78	5.7	204	3.6	1.92	4.04	15	172	30
6	58.4	61.6	52.1	58.5	1.71	1.62	2.78	5.8	208	3.6	1.92	4.04	15	175	30
7	50.2	55.1	42.7	51.5	1.99	1.81	3.60	7.0	194	4.2	2.34	4.82	10	175	25
8	50.2	55.1	42.7	51.5	1.99	1.81	3.60	7.6	211	4.2	2.34	4.82	8	175	23
9	50.2	55.1	42.7	51.5	1.99	1.81	3.60	7.8	217	4.2	2.34	4.82	8	180	23

the variation in the results given by these rules, and emphasizes the need of a perfect understanding of the wide difference between the theoretical and practical operation of compound locomotives

The important fact is shown that the ratio of the initial and final volumes in nowise indicates the real ratio of expansion. This results from the effect of the comparatively small receivers used, which gives a large drop in pressure in the receiver, 25–26, and the wire-drawing due to inadequate valve motion. This more particularly applies to the conditions when the engine is running at considerable speed, for at such times the reduction of pressure due to wire-drawing is equal to or greater than the reduction resulting from expansion. This shows how a compound locomotive at high speed may approach more nearly to a throttle-governed wire-drawing steam engine than to one having a variable cut-off. It is this action which reduces so greatly the otherwise possible saving of a compound locomotive at high speed, 139–147, and when taken together with losses resulting from compression gives nearly a full explanation of the reasons why a majority of compound locomotives, thus far, have not shown a very substantial saving in passenger service.

Note.—The terminal pressure for Table K is not taken as that at exhaust, but is taken at an equated pressure lying between that at exhaust and that at the end of the stroke. This is done to allow for the useful work done by the steam during exhaust before the end of the stroke is reached. The equated terminal pressure thus taken is not an arithmetical mean of the pressure at exhaust and the pressure at the end of the stroke, but is so selected as to allow for the work done from the exhaust point to the end of the stroke. For the slow-speed cards it is taken nearly at the exhaust point, and for the high-speed cards nearly at the end of the stroke.

47. **Ratio of Cylinders — Elementary Formulas for.** — In treatises on compound engines, formulas have been deduced for the ratio of the volumes of the cylinders so that the total work, 22, done by the engine will be almost equally divided between the cylinders, 15, but such formulas are not applicable to engines having much compression and wire-drawing, and therefore not to locomotives. Usually for engines with receivers, these formulas are based upon a constant receiver pressure. A rule that has been frequently used is that the ratio of the volumes of the two cylinders should equal the square root of the total number of expansions desired. This rule will not apply to locomotives.

48. **Ratio of Cylinders as Affected by Maximum Width of Locomotive.**—So far as economy alone is concerned, the maximum over-all-width of a locomotive and the necessity for a minimum weight of reciprocating parts places such a low maximum limit upon the diameter of the l. p. cylinder of a two-cylinder receiver compound locomotive that it cannot always be given a volume that will give the best theoretical economy; however, single expansion locomotives frequently work with such low efficiency that a compound can generally be given sufficient volume in the l. p. cylinder to enable it to show a substantial saving, although not the maximum saving that would be possible under other and more favorable conditions. The loss due to existing types of valve motion is so great that the comparatively minor loss incident to a reasonable limitation of the diameter of the l. p. cylinder practically disappears in comparison. With the four-cylinder compound, it is possible to get a more economical volume of l. p. cylinder, but it would appear from what has been done so far that the four-cylinder compound introduces further troubles in the valve motion, and the saving that would otherwise be found, by reason of the larger l. p. cylinder, is to some

extent counterbalanced by a decrease in the efficiency of the valve motion. This more particularly applies to high speed locomotives.

It should be mentioned here that the use of a double l. p. cylinder, as originally proposed by Mallet, 112, see Figs. 28 and 100, will give a sufficiently large l. p. cylinder capacity to any compound locomotive of the two-cylinder type, 51.

The term "valve motion," as here used, refers to sizes of ports and all parts of the steam regulating gear.

49. Ratios of Cylinders Commonly Used.—The cylinder ratios, which have been used for two-cylinder compounds, range from 2.74 for small engines to 1.77 for large engines. Four-cylinder engines generally have a ratio of about 3. Mr. Mallet, from his wide experience, has said that the ratio for two-cylinder engines should not be less than 2. Mr. von Borries recommends ratios of 2 for freight locomotives and 2.25 for passenger locomotives. Ratios between the limits of 2, and 2.2 have been adopted by the majority of designers. For two-cylinder compounds in the United States a ratio of 2.1 has been more generally used. The foregoing gives prevailing practice.

As has been shown, mathematical calculation is not of much value in determining the cylinder ratios, when such calculation is based upon the elementary engine. It has also been shown that the relative mean effective pressures in the cylinders is more dependent upon the valve arrangement at the present time than upon the sizes of the cylinders, and, therefore, as the power in the cylinders depends upon the mean effective pressure, it follows that the division of power between the cylinders depends not so much on the sizes of the cylinders as upon the action of the valve motion and the size of the steam passages. And, further, the necessity for starting trains quickly is such a controlling condition that the ratio of the cylinder volumes must be made, not what is most efficient from an economical stand-point, but rather what will give a reasonably uniform power at starting, and sufficient power on the h. p. side to enable the engine to start without pulsations and jerks.

It is evident that the best practical ratio of cylinder volumes must have been originally determined by experiment. Experiments in cylinder ratios have been made by nearly all who have undertaken to introduce two-cylinder compounds, and many have traveled over the ground of investigation covered by others, with the hope of getting a satisfactory starting power and an even power distribution with better theoretical conditions for economy In a recent case of this kind, the locomotive builder had to take off the h. p. cylinder and replace it with a larger one. The ratio at first was about 3, and it was finally made about 2.2 to 1.

To emphasize and explain what has been sa d regard.ng the incidental control of the mean effective pressures in the h. p. and l. p. cylinders by the wire-drawing and compression, reference is now made again to Figs. 14 and 15, Cards 1 to 9. See also Tables C, D, E, F, G, H, and I. These cards represent about the best that has been done in the way of an equal distribution of power between the h. p. and l. p. cylinders of large two-cylinder compound locomotives.

50. **Ratio of Cylinders as Affecting Equalization of Power in Two-Cylinder Receiver Compounds.**—Theoretical investigation has had but little to do with developing the proper ratio, but practical experiment has shown definitely that a ratio of 2.4 is as great as can satisfactorily be used in a two-cylinder compound, and that a ratio of 2 is better, as it makes easier the approximately equal distribution of power between the cylinders at different speeds and gives better results in starting heavy trains. It is a simple matter to adjust the equalization of the power in the cylinders of a two-cylinder receiver compound with a

volume ratio of 2 when the valve motion is good. It is easier to accomplish this equalization with a ratio of 2 than with a ratio of 2.4. With a ratio of 2.4 it is practically impossible to equalize the power between the cylinders at high speed, unless the ports and passages are unusually large and the valve motion most excellent.

All things considered, it is better to assume the ratio of volumes of cylinders for two-cylinder receiver compound locomotives between the limit of 2 and 2.2, than to go outside of these limits with the hope of obtaining greater economy. Within these limits, it does not matter so very much what the ratio is; but, as has been said before, it is easier to adjust the equalization of power between cylinders, particularly for high-speed work, when the lower limit is used, and in addition better results will be obtained in starting trains.

Exact equalization of power is not necessary, or perhaps desirable. A variation of 10 per cent. either way will produce no harmful results. In the case of some recent two-cylinder receiver compounds, the greatest variation in power from starting to a speed of 67 miles per hour is 5 per cent. This is a remarkably close equalization.

51. Ratio of Cylinders and Equalization of Power in Non-Receiver Compounds.— For four-cylinder receiver or non-receiver compounds having duplicate sets of cylinders on the two sides, where the equalization of power is not so desirable as in two-cylinder receiver compound locomotives, a ratio of from 2.7 to 3.2, as limits, can be chosen without error and without materially affecting the economy in locomotive work. Probably a ratio of 3, for the present at least, will be found perfectly satisfactory.

If the time ever comes when a better positive acting valve motion is devised, 8, 82, and one that will, with the assistance of larger valves and steam passages, give quicker and greater port openings and will postpone the point of compression nearer to the end of the stroke, then these remarks about the cylinder ratios for compound locomotives will perhaps need to be modified ; but until then the limits of ratio given will be found satisfactory.

52. **Ratio of Cylinder Volumes to the Work to be Done.**—The ratio of the cylinder volumes, not to each other but to the work to be done, is an important matter. In general, in this country, the two-cylinder receiver compounds have had less volume than they should have for the work they have been designed to do. This has perhaps been caused by the timidity with which designers have undertaken larger cylinders with their consequent heavier reciprocating parts for American engines. The cylinder volumes used in Europe for the same work are greater in proportion to the hauling power of the locomotive, as determined from the total weight on drivers, than they are here, Table L, Appendix Q. On the other hand, the four-cylinder non-receiver engines built here have had cylinder volumes more in proportion for the work to be done, and more in accordance with European practice. This appears from Table L, which gives the comparative cylinder volumes of several designs. An increase of total cylinder volume for two-cylinder compounds above that now generally used in this country is certainly necessary if the best attainable efficiency is sought.

For the Vauclain compound the Baldwin Locomotive Works have used the following formula for a number of engines, but at the present time they are using a formula having a somewhat different coefficient, instead of 2.7, and this gives larger cylinders for the same weight of locomotive:

$$d'^2 = \frac{D.}{2.7\,PS.}$$

$$d^2 = \tfrac{1}{3}\,d'^2.$$

In these formulas the following are the meanings of the symbols used:

P = Pressure, by gauge, at admission to h. p. cylinder.
S = Stroke in inches.
D = Diameter of drivers in inches.
W = Weight on drivers in pounds.
d = Diameter of h. p. cylinder in inches.
d′ = Diameter of l. p. cylinder in inches.

Mr. von Borries has recently said that, in his opinion, at the present time the following proportions should be used:

Cylinders.—Diameter d of l. p. cylinder to be calculated by the formula

$$d^2 = \frac{4\ T.\ D.}{p.\ s.}$$

if the full tractive force is to be used as in ordinary goods engines. In this formula is:

T = Tractive force = $\frac{18}{100}$ of adhesive weight.
D = Diameter of driving wheels.
p = Boiler pressure.
s = Stroke of pistons.

For passenger and fast-traffic engines, where calculation is difficult, the diameter of l. p. cylinder of compound engines to be 1½ the diameter of cylinders of single expansion engines, raising the steam pressure at least 15 pounds.

Diameter of h. p. cylinder to be 0.7 of l. p.

Receiver.—The volume must not be smaller than h. p. cylinder, better 1.50 of this.

Ports.—The dimensions of ports are shown in Table M.

TABLE M.

	H. p. cylinder.	L. p. cylinder.
Clearance (including ports), - - -	0.05	0.07 of volume of l. p. cyl.
Area of ports, - - - - - - -	0.04	0.07 of area of l. p. cyl.
Width of ports, - - - - - - -	0.056	0.07 diameter of l. p. cyl.
Length of ports, - - - - - - -	0.56	0.77 " " "

For freight engines dimensions of ports can be 5 per cent. smaller.

Motion and Slide-Valves.—If t, is the width of l. p. steam-port the following proportions should be used:

Travel of valves for middle position of link, - - - - - - 1.6 .t
Outside lap of both valves, - - - - - - - - - 0.7 t.

Inside clearance of h. p. valve, - - - - - - - - - - 0.20 t.
Inside clearance of l. p. valve, - - - - - - - - - 0.

The corresponding sections of slide-valves and faces are shown in Fig. 27. The dimensions are given in proportion to *t* as a unit.

The link-hanging rods to be made of different length, so that 0.4 cut-off in h. p. cylinder corresponds to 0.5 in l. p. cylinder.

Greatest cut-off running forward to be 0.77 in h. p. and 0.8 in l. p. cylinder.

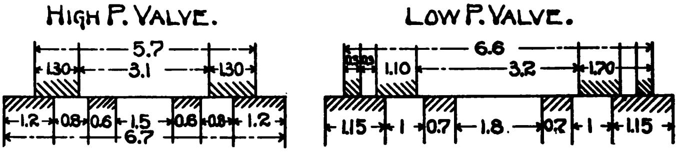


FIG. 27.
von Borries' Proportions of Valve Dimensions.

Mr. A. Mallet and Mr. A. Brunner have found from experience that a ratio of 2.25 is preferred to any other for cylinders of two-cylinder receiver compounds. With

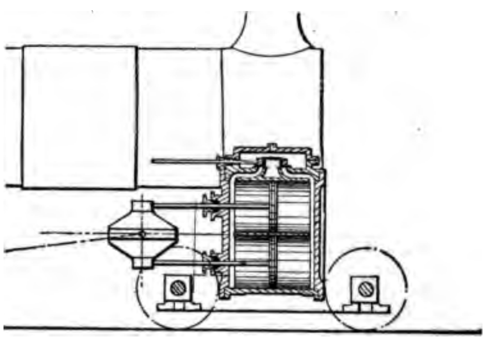


FIG. 28.
Lapage Double Cylinder.

this ratio these designers have used the same cut-off in both cylinders. With a ratio of 2 a longer cut-off is needed in the l. p. cylinder.

It would seem that the proposition of Mr. Mallet, and later by Mr. R. H. Lapage, to use a double l. p. cylinder, as shown by Figs. 28, 29 and 100, would effectually dispose of the problem of finding room for a large l. p. cylinder.

When this double cylinder is used in conjunction with a crosshead of the Vauclain type, shown in Fig. 119, it is not clear why a two-cylinder receiver compound, if such it could then be called, having in reality three cylinders,

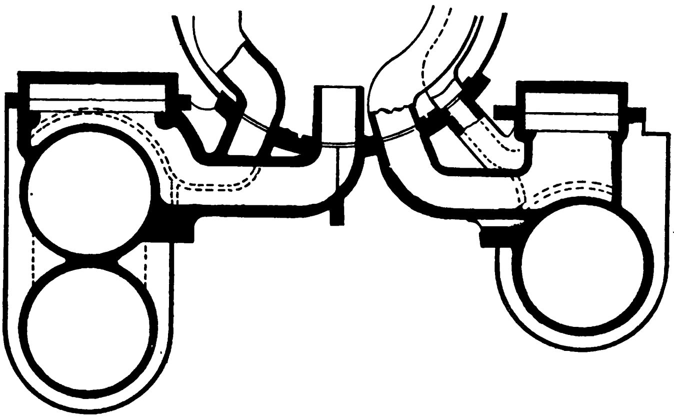


FIG. 29.
Lapage Double Cylinder.

could not be built with sufficient cylinder capacity for any of the largest locomotives now made. This proposition has considerable merit, and if two-cylinder compounds with receivers are continued in use, and there is much prospect that they will be, it is probable that some extended practical use will be made of this suggestion.

CHAPTER VIII.

RECEIVER CAPACITY, RE-HEATING AND SEQUENCE OF CRANKS.

53. **Receiver Capacity.**—The capacity of a receiver can be properly based on the capacity of the h. p. cylinder. In general, the greater the capacity of the receiver the more readily can the equalization of power between the two cylinders be accomplished by an adjustment of the cut-off, 22, in the cylinders, and the less will be the effect of a change in the sequence of the cranks. Large receiver capacities give less variation of pressure in the receiver, and in this way are conducive to economy. The ratio of the receiver volume to the volume of the h. p. cylinder now commonly used for locomotives is given in Table U1. Probably in no case is it advisable to use a receiver with less capacity than 2.3 times the volume of the h. p. cylinder, and it is better to use a higher ratio. Some successful four-cylinder receiver compounds have a receiver volume 4½ times the volume of the h. p. cylinder. The prevailing practice here is shown by Table U1. For comfortable working the volume of the receiver should not be less than 2.5 times the volume of the h. p. cylinder.

Mr. A. Brunner, who has made many designs of compound locomotives for Mr. Mallet, is of the opinion that the receiver should have from 4 to 5 times the volume of the h. p. cylinder.

54. **Re-Heating and Steam Jackets.**—The receivers should be located in as hot a place as possible; not so much to gain re-evaporation or super-heat as to prevent condensation. If the receiver is exposed to the atmos-

phere, the condensation in cold weather would be so enormous as to offset any possible saving from compounding. There is no doubt but that some re-evaporation of the moisture in the steam does take place in the receiver of a compound locomotive when the receiver is in the smoke box, more particularly when the engine has short tubes and is working hard, as on a grade, or whenever the conditions are such as to give a high smoke box temperature; but there is probably no material saving in present designs of compound locomotives over single expansion engines that results from re-evaporation in the receiver. The re-heating must be small owing to the short time, about one-third to one-fifth of a second, that the steam is in the receiver when the engine is at speed. However, all that is gained by re-evaporation is purely a saving, for the smoke box heat which produces the re-evaporation would otherwise be wasted through the stack. If a steam jacket is used on the receiver, or on either of the cylinders, the steam used in it for re-heating would be used in the cylinders if there were no jackets, and therefore the saving in the cylinders from a steam jacket is offset by the loss of the steam used in the jacket. Mr. F. W. Dean has tried a steam jacket on a two-cylinder receiver compound locomotive for the Old Colony Road, but it was finally abandoned on account of the difficulty of draining it, and the engine now runs without the steam jacket. The space in the jacket now serves to give better heat insulation to the h. p. cylinder on which the jacket is placed.

As it does not matter much in a compound engine whether the jacket is on the receiver or the h. p. cylinder, it is probably better, if a steam jacket is wanted, to put the receiver into the boiler itself, as has been done on a Lindner compound in Germany. This plan removes any difficulty of draining the jacket and gives the highest possible value to steam jacketing. However, as has been said, the re-heating in the receiver brought about by a steam jacket is not all gain, as there is some loss of steam in the jacket or in the boiler as the case may be; but with re-heating by the smoke box gases, all re-heating is purely gain. It is probable that such gain as is obtained from re-evaporation in a receiver in a locomotive smoke box, under ordinary conditions, is greater than could possibly be obtained from a steam jacket on either the receiver or the h. p. cylinder. It is now generally understood that a steam jacket on the l. p. cylinder is not conducive to economy.

In order to gain all that is possible by a re-evaporation in the receiver produced by the heat in the smoke box gases, it is better to use a large receiver made of one or more copper pipes. It seems impractical to put these pipes in the hottest part of the smoke box; namely, in front of the tubes, because of the difficulty in reaching the tubes for cleaning and repairing; hence, it is customary to put the receiver pipe around the top of the smoke box, either forward or back of the smoke-stack opening.

Cast iron receivers have been used generally in this country. They cost less and have greater durability than copper. It is not now known whether the thin copper receiver gives a compound locomotive greater efficiency than a cast iron receiver.

55. **Smoke Box Temperatures.**—Smoke box temperatures vary from 400 to 1,200 degrees, according to the forcing of the engine and the length of the tubes. Recently there has been a decrease in smoke box temperatures with new designs of locomotives, resulting from the use of larger fireboxes and longer tubes, and it is probable that smoke boxes will be run at a lower temperature in the future than they now are, but in no case will they reach so low a temperature as to remove all value for the purpose of re-evaporating moisture in the steam in the receiver of two-cylinder receiver compound locomotives.

56. Sequence of Cranks.—At the commencement of the use of compound cylinders for locomotives it was questioned whether the h. p. or the l. p. crank should precede in rotation, but as soon as the receiver capacities were made sufficient, it was found that there was little or no difference which crank had precedence in receiver engines. For non-receiver engines it would make quite a difference which crank precedes if the cranks were placed at an angle with each other, but as non-receiver compounds for locomotives are only made with the h. p. and l. p. pistons connected to the same crank, it is not necessary to discuss this special case. Practically, the sequence of cranks need not enter as a problem for solution in compound locomotive designing.

CHAPTER IX.

MAXIMUM STARTING POWER OF LOCOMOTIVES.

57. **Starting with Close Coupled Cars and with Free Slack.**—In starting a train it makes considerable difference whether the train is close coupled, like a vestibuled passenger train, or has free slack as with a link and pin coupling. With a close coupled train it is more difficult, as the locomotive can only move forward a very short distance before the entire load has to be started, whereas with free slack the locomotive can frequently move a full revolution before taking up the last car. For this reason compound locomotives have given more trouble in starting passenger trains than freight trains.

58. **Starting of Two-Cylinder Receiver Compounds without an Independent Exhaust for High-Pressure Cylinder.**—Two-cylinder compounds can generally accelerate passenger trains without difficulty, but there are certain positions of the cranks in which such locomotives have a reduced power, and when in such position the two-cylinder compound of this type does not accelerate trains, either passenger or freight, as satisfactorily as the ordinary or single expansion engine. The reason is, that while the maximum turning moment of a compound locomotive at starting, which occurs when the l. p. crank is nearly on the quarter, is greater than the starting power of a single expansion engine as a rule, yet the minimum starting power, which occurs when the h. p. crank is about on a quarter, is considerably less than with the single expansion engine. This result comes from the comparative size of the h p. cylinder, it being but little if any larger than one cylinder of a single

expansion locomotive, and yet has a back pressure on one side of the piston very nearly equal to one-half the boiler pressure, whereas the single expansion cylinder has but a very small back pressure. Hence, while the compound has, perhaps, 10 per cent. larger cylinder, it has fully 40 per cent. less effective pressure. This is probably all the argument that is necessary to show why it is that the practical conditions of operation compel the use of a larger cylinder on the h. p. side of the two-cylinder compound than is generally used for a single expansion engine. See Chapter XVII for argument about recent tendency in starting gears.

59. **Starting of Two-Cylinder Receiver Compounds with Independent Exhaust for High-Pressure Cylinder.**—The engines of this class start and accelerate trains equally as well as single expansion locomotives, and are practically such at low speeds when the separate exhaust is opened. At higher speeds, the small opening allowed for the separate exhaust generally causes considerable back pressure, and the engine will not work well with single expansion for that reason. This class of compounds can generally start heavier trains than single expansion locomotives of equal rating, for the reason that the cylinders are larger; but, of course, the limit of all traction engines lies in the adhesion of the drivers to the rails; hence, the additional cylinder power of this type of compound is of no advantage after the limit of adhesion is reached.

60. **Starting of Four-Cylinder Two-Crank Receiver and Non-Receiver Compounds.**—The four-cylinder two-crank compounds do not have the disadvantage common with two-cylinder compounds without separate exhaust for the h. p. cylinder, at starting, as live steam can be used in both l. p. cylinders, one on each side, and the engine can be started under a heavier load than it can haul under normal conditions of compound working. Generally speaking, four-cylinder two-crank compounds have more

starting power and more ultimate hauling power than single expansion locomotives of equal rating. This applies to four-cylinder tandem receiver compounds and all four-cylinder compounds having but two cranks. This increase of hauling power is one of the strong claims made by the advocates of four-cylinder two-crank compounds. In cases where it is customary for single expansion engines to separate trains in two parts and pull each part separately over a heavy grade, joining the train together again on the other side, generally called "doubling the hill," the four-cylinder two-crank compound and the two-cylinder receiver compound having independent exhaust to the open air for the h. p. cylinder, can be made to haul the entire train over the hill by using steam directly from the boiler into the l. p. cylinders and running the train at a comparatively low speed. This is certainly a decided advantage on some roads.

61. **Starting of Four-Cylinder Four-Crank Compounds with Receivers.**—The starting power of four-cylinder four-crank compounds depends upon the location of the cranks, and whether parallel rods are used. With some of these types the starting power has been small; with others it has been ample, 128–134. See Appendix K.

62. **Starting and Hauling Power of Single Expansion Locomotives.**—The following formula has been much used for the tractive power of locomotives:

$$T = \frac{d^2 p s}{D}$$

in which d = the diameter of the cylinders in inches, p = the mean effective pressure in pounds per square inch, s = the stroke in inches, D = the diameter of the driving wheels in inches, and T = the tractive power or pull at the rail in pounds. This formula is based upon the fact, that, neglecting friction, the work done in both cylinders during any period, such as one revolution, is equal to that done at the circumference of the driving wheel during the same time. It is convenient and practical, as it gives the hauling power of the locomotive when the mean effective pressure in the cylinders is known. The tractive power by this formula includes the power necessary to move the entire mechanism of the locomotive and the locomotive itself. It is, in fact, the entire work done in the cylinders reduced to an equivalent pull on the rail. In using it, a deduction must always be made for the internal friction of the engine and for the power required to move the engine and tender in order that the actual pull on the train itself may be determined. Some have made the error of assuming a universal value for p, namely, 85 per cent. of the boiler pressure. This is greatly in error when applied to some engines, and the only safe way to use the formula for a given engine is to determine, by taking indicator cards from the engine in question or a similar one, what is the real maximum mean effective pressure. The method of deducing this formula will be found in Appendix H. It follows from the method of deduction that this formula gives an average value for the pulling power, and therefore that, while it furnishes a ready method of comparing the pulling power of locomotives under ordinary conditions, it is of very little use in estimating the first starting power from a stand-still, since the *minimum* pull, and not the *average*, is the practical measure of the initial starting power of the locomotive.

In the single expansion locomotive, assuming that steam can be admitted during the full stroke, and neglecting the effect of angularity of connecting rods, the minimum pull occurs when one crank is on the half centre, the other being at a dead point, and the maximum pull is developed when both cranks make an angle of 45 degrees with the centre line through the dead points. This can be readily demonstrated by calculation, or by a graphical construction.

63. Graphical Representation of Hauling Power.—There are several methods of representing rotative efforts graphically, one of which is shown by Fig. 30, in which the dotted line *a* . . *a* represents the rotative effort, or the tangential pull or push, on one crank pin, and *b* . . *b* is that of

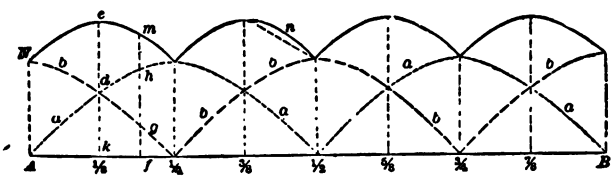


FIG. 30.
Diagram Showing Combined Starting Power of Both Cylinders of a Single Expansion Locomotive.

the other at right angles to it, the steam pressure being assumed as constant throughout the stroke.

The method of construction is as follows: Let *A B* be the length of the circumference of a circle, of which *C D*, Fig. 31, is the radius. It can be readily shown that the component *D F*, of the pressure on the piston *D H*, which tends to produce rotation, is proportional to the sine of the angle *a*, through which the crank has turned from a dead point. Divide the line *A B* and the circumference in Fig. 31 into the same number of equal parts. Then through the points of division on *A B* lay off perpendicular distances, such as *k d*, equal to the lines which represent the sines of the angles in Fig. 31, such as *K D*.

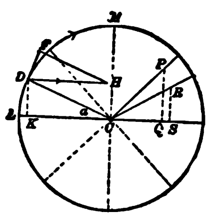


FIG. 31.

The dotted curve *a a* represents the variations in rotative efforts on the crank starting from *C L* during one revolution, and the curve *b b*, shown by a broken line, represents the variations in efforts on the crank starting at *C M*, or at right angles with the first.

The total rotative effort is shown by the ordinates of the full line curve in Fig. 30, which is obtained by adding the ordinates of the curves for the single crank, for example, $fm = fg + fh$. It is evident that the value of the total efforts varies between *A N* and *k e*. In the first case, one crank is on the dead point, and the other is on the half centre, or midway between the two dead points. The pull at the rail is then:

$$\tfrac{1}{4} \pi d^2 \times p \times s \div D,$$

Which is .7854 of the tractive power as found by the ordinary formula. In the second case the pull is twice that of one crank when making an angle of 45 degrees with the centre line, or it is

$$\tfrac{1}{4} \pi d^2 \times p \times 2 \times .707 s \text{ , } \div D,$$

Which is 1.11 of the tractive power as usually estimated.

It is also clear that there are four maximum and four minimum points during a revolution. These values are determined as has been said, on the basis that a constant steam pressure can be maintained throughout the stroke, which would be the case in starting if steam could be admitted to the cylinder during the whole stroke. But when the latest cut-off takes place, when the piston is some distance from the end of the stroke, as, for example, at 21 inches with 24 inches stroke, the engine will have a weaker position for starting than that given above as a minimum. When one piston is 21 inches from the beginning of its stroke the other will be about 4 inches from the beginning of its stroke, and its crank will have turned through about 50 degrees from a dead point. If cut-off takes place at 21 inches, no steam can be admitted to that cylinder during the remainder of the stroke, that is, if the start occurs with the piston in this position, and the work of starting devolves upon the other cylinder.

When the piston has moved 4 inches from the beginning of the stroke the rotative effort is about ¾ of the maximum for one cylinder, and is, therefore, about .589 of the tractive power as usually estimated. This corresponds to an ordinate of the curve *a a*, a little to the right of *k d*, and is evidently the most difficult position from which to start the single expansion locomotive. The reduction in the rotative effort on account of the fall in pressure due to the expansion after cut-off and release will be slight. This can be shown on the diagram by laying off radial distances such as *C P* and *C R* on the proper radii to represent the pressures for these crank positions, and using the lines *P Q* and *R S* for ordinates in Fig. 30, instead of those used before. The final effect is shown by the dotted curve at *n*, Fig. 30.

As the locomotive starts the mean effective pressure in the cylinders will be somewhat reduced, but the reduction will not be of large amount within what may be called the starting limits, or until the link would ordinarily be hooked up. As the speed increases the inertia of the reciprocating parts, etc., will be sufficient to modify the form of the diagram of crank efforts, but it is not necessary to consider that in estimating the initial starting power.

64. **Starting Power with Mallet's System and other Non-Automatic Starting Gears.**—Turning now to the compound locomotive, it is apparent that in the Mallet and other systems having independent exhaust for the h. p. cylinder the starting conditions are almost identical with those in the single expansion locomotive. If the h. p. cylinder is of the same size as one cylinder of the single expansion locomotive, and the cylinder ratio is 2, it is only necessary to admit steam of one-half the boiler pressure to the l. p. cylinder in order to have starting power equivalent to that of the single expansion engine, the same boiler pressure being used. If the l. p. initial pressure is greater than one-half the boiler pressure, the starting power of the compound will be greater than that of the single expansion engine in all positions in which the l. p. cylinder is available for use in starting, that is, except when the l. p.

crank is on a dead point, or when the l. p. valve is in such a position that steam cannot be admitted. If the boiler pressure of the compound is higher than that of the single expansion engine, and the h. p. cylinder is the same size as one of those of the single expansion engine, the starting power of the compound engine of this type will be the greater in about the proportion of the two boiler pressures.

65. **Starting Power with Worsdell, von Borries and other Automatic Starting Gears.**—In the Worsdell and von Borries type, and others with automatic intercepting valves, the conditions in starting are quite different from those just described. When steam is admitted to the

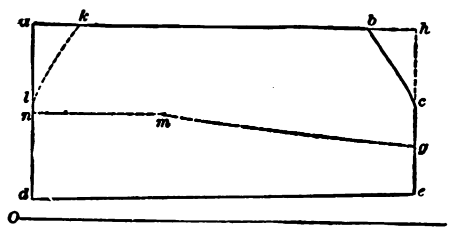


FIG. 32.

Steam Pressure During First Revolution with an Automatic Starting Gear.

receiver by means of the starting valve, the intercepting valve is closed, and the h. p. piston therefore starts against the pressure of the steam or air which filled the receiver just before the starting valve was opened. The amount of this receiver pressure will depend upon the length of time during which the engine has been standing, the condition of the valves, etc. If at starting the h. p. crank is at a dead point, the pencil of an indicator, which is applied to the steam end of the h. p. cylinder during the first stroke, will trace a line similar to *a b c*, Fig. 32. The back pressure acting against the other side of the piston during this stroke is shown by a line such as *d e*, the pressure at *e* being somewhat greater than that at *d* on account of the compression in the h. p. cylinder and receiver. The initial back pressure is assumed in the present case as equal to the atmospheric pressure. The diagram, *a b c e d*, thus represents what may be called the effective indicator card for the first stroke of the h. p. piston.

When the h. p. exhaust opens the pressure in that cylinder and the receiver will fall to some point *g*, which can be only approximately determined by calculation. It is located on Fig. 32, by calculation on the basis of no condensation or evaporation during the exhaust. The forward pressure on the h. p. piston during the second stroke will be similar to that during the first stroke, and is shown in Fig. 32 by *h k l*. The back pressure line during this stroke will consist of, first, a curve *g m*, which represents the compression by the h. p. piston of the steam which fills the space between the h. p. piston and the intercepting valve, until that valve opens; and second, of a line *m n*, of nearly constant pressure, which represents the back pressure during the remainder of the stroke, after the intercepting valve opens and the starting valve is closed.

It is generally assumed that the pressure of the steam, which is admitted directly to the receiver in starting, is reduced by wire-drawing to about one-half the boiler pressure. Assuming this to be so, the h. p. cylinder back pressure will become sufficient to open the intercepting valve when about 5/8 of the second stroke has been accomplished, as indicated at *m*, Fig. 32. The net diagram from which the effective pressure on the h. p. piston for the second stroke can be obtained is then *h k l n m g*.

A diagram of rotative efforts constructed from these indicator cards is shown in Fig. 33 by the curve *A E C F B*, from which the reduced effort resulting from the increasing back pressure during the second stroke is apparent.

The distribution of work in the l. p. cylinder in starting does not differ from that in the single expansion engine.

The rotative effort will, therefore, be represented by a curve such as *H K L D M*, Fig. 33, which has the same form as the single crank curves in Fig. 30. The curve in Fig. 33 is constructed on the basis of the initial l. p. pressure, being one-half of the boiler pressure. If the initial pressure is greater than this, the ordinates of the curve between *H* and *K*, *K* and *D*, etc., should be proportionately increased. The combined effort of the two cylinders is shown in Fig 33 by the full line curve. The intercepting

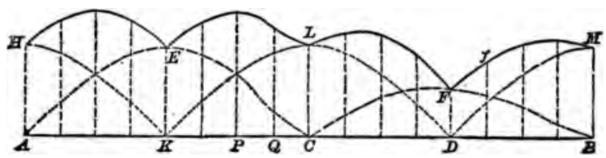


FIG. 33.
Starting Power During First Revolution of a Compound, with Automatic Starting Gear.

valve opens at about the point *f*, and from that point the engine will work as a compound.

It has been already shown that when so working with the customary pressures the power developed at late cut-offs is less than that of the single expansion engine. The location of the point at which the intercepting valve opens depends upon the pressure in the receiver before starting, the pressure of the steam admitted to the receiver by means of the starting valve, and the size and location of the receiver. For any given combination of conditions it will be found at a definite distance from the point *C*, or from the end of the first stroke of the h. p. piston. In the present case this point was found to be about 5/8 of the stroke from *C*.

It is obvious that this action is not at all dependent upon t e first stroke of the h. p. piston, but only upon the exhaust from that cylinder. It follows from these considerations that, if the h. p. crank is at a dead point at starting, the engine will move through something over 3/4 of a revolution before compound working begins: but, on the other hand, if the h. p. piston is at the position corresponding to *P*, or near the point at which cut-off takes place, the compound working will begin after about $\frac{7}{16}$ of a revolution. If the h. p. crank is in some position such as *Q*, at which the steam valve is closed, the starting must be accomplished by the l. p. cylinder alone; but after a slight movement, sufficient to carry the h. p. crank over the dead point, the cycle will continue as if started at *A*, the effect being to prolong the time of direct working of the l. p. cylinder to about $\frac{7}{8}$ of a revolution.

After compound working commences, and while admitting steam for as much of the stroke as possible, the combined diagram of rotative efforts would be similar to Fig. 30, but with a smaller mean effective pressure, the proportion being, with boiler pressures of 170 and 150 pounds in the two types, not greater than 110 to 122, as has been already mentioned. The two diagrams, Figs. 30 and 33, are not drawn to the same scale of pressures, but the shape of the full line curves represents with reasonable accuracy the variations in starting power in the single expansion and compound locomotives. In conclusion, it appears that, with the pressure customary in the two forms, the pulling power of the Worsdell and von Borries type, and others with automatic intercepting valves, in starting may be greater than that of the single expansion engine having cylinders of the same size as the h. p. cylinder, during the first half revolution approximately, but that after this the power of the compound engine diminishes until it is from 80 to 85 per cent. of that of the single expansion engine.

66. **Starting Power with the Lindner System.**—The maximum starting power of the two-cylinder Lindner type with latest type of Lindner starting gear, and without intercepting valve, is about the same as the maximum with the two-cylinder type having automatic intercepting valves, but

is much less than the two-cylinder type having independent exhaust for the h. p. cylinder. Appendix L gives analysis of the starting power of a Lindner engine.

67. **Starting Power of Three-Cylinder Three-Crank Compounds.**—The starting power of three-cylinder compounds, when the drivers are coupled together with parallel rods, is about the same as with the two-cylinder type. If the drivers are not coupled, as with the Webb type, the ultimate starting power is dependent upon the accidental location of the crank at the time of starting. The minimum starting power for full cut-off and full throttle of a three-cylinder type without parallel rods is lower than the minimum of the two-cylinder receiver type. See Appendix I.

68. **Variation of Hauling Power with Four-Cylinder Two-Crank Receiver and Non-Receiver Compounds.**—The curve of variation of hauling power during a complete revolution in a two-crank four-cylinder non-receiver compound or four-cylinder tandem receiver compound, does not differ materially from that of a single expansion engine, as both sides are identical in action. However, with the same number of expansions in the four-cylinder and the single-expansion engine, the hauling power is more uniform in the compound, more particularly for the reason that the cut-off in both cylinders in the compound is later than in the single expansion engine. In any engine, the later the cut-off the more uniform will be the hauling power during a complete revolution at slow speed. At high speeds this is much modified by the inertia of the reciprocating parts, see Appendix P. Uniformity of pull on a train is of more importance in starting and at slow speed than at high speed, as at slow speed a variation in the pulling power may be felt by the passengers in a train.

The foregoing statements about four-cylinder compounds apply more particularly to the non-receiver Vauclain, Johnstone, and tandem types, and to the tandem form generally, but are also true of four-cylinder receiver compounds with four cranks in which the cranks are almost evenly divided in position on a circle and with parallel rods between the axles having cranks.

CHAPTER X.

CONDENSATION IN CYLINDERS.

69. **Range of Temperature.**—When compound engines are well designed and are working under favorable conditions, the loss from condensation of steam in the cylinders should be less than with single expansion. This arises from the lower range of temperature in the cylinders; the range of pressure being less in the cylinders of the compound, it follows that the range of temperature would also be less. However, the gain in efficiency by saving in condensation may be more than offset by results of faulty mechanical arrangement. If the cylinders, steam passages, and receiver, are not well protected from radiation, the loss by condensation from this cause may more than offset the saving from the reduction of condensation brought about by a lower range of temperature in the compound cylinders.

70. **Need of Covering Hot Surfaces to Prevent Radiation.**—It is a very bad, but common practice, in locomotive construction, and one that has descended from the past, to construct cylinders for locomotives with the walls of the steam passages exposed directly to the atmosphere without covering on the outside. Steam chests and cylinder heads are likewise very poorly insulated in common practice. The loss from these defects alone is so great that it is hardly worth while to go to the trouble to use compound cylinders unless the heat insulation is improved. This common defect in locomotive construction has been the subject of severe criticism by mechanical engineers who are familiar with the better class of designing for marine and stationary engines. Just now some railroad companies have

taken the matter in hand and are using somewhat better heat insulation for all parts that are exposed. Locomotive builders, however, have not yet considered it worth while to reduce radiation by better insulation, probably because of the lack of appreciation of these losses on the part of those who purchase locomotives. Mr. F. W. Dean, **54**, in designing some engines for the Old Colony Railroad, has separated the steam pipes from the walls of the cylinders, and has used a better degree of heat insulation than is common. From the results obtained from his engine, it would appear that the better insulation has been of a decided advantage. The condensation of steam in a locomotive is one of the sources of loss, and the highest possible saving of the compound cannot be obtained without a proper insulation of all pipes, passages, and receptacles for steam.

71. **Condensation, Leakage of Valves and Re-Evaporation as Determined from Indicator Cards.**—In the discussion of a method of analysis of combined indicator cards, the losses due to condensation are considered, **42, 44.** In addition to that discussion, the following further analysis of Fig. 26, and some cards from other types of engines, will be found instructive. This analysis shows how the steam weights calculated from actual indicator cards vary at different points during a stroke in the h. p. and l. p. cylinders of two-cylinder receiver and four-cylinder non-receiver compounds. These results are given in Tables N, J and O. Table N gives the fundamental data regarding the engines that is used to make the calculations from the indicator cards. Table J gives the final results of the calculation, and shows the weight of steam used per stroke in both cylinders, and the per cent. of increase or decrease of weight of steam used in the l. p. cylinder above or below that used in the h. p. cylinder, this data being taken from the measurements on the indicator cards. The

TABLE N.

Giving Details of Calculations for Table J. Also, giving Dimensions of Locomotives Referred to in Table J.

	Baldwin 10 wheel, No. 82 Vauclain Comp. Card No. 3.	C., B. & Q. Mogul, No. 324. 2-Cylinder Comp. Card No. 49.	Schenectady 12 wheel Cards 8 & 8a.	Baldwin 10 W. Comp. No. 82. Vauclain. Card No. 51.	Baldwin No. 82 on C., B. & Q. Card No. 33.	C., B. & Q. Mogul, No. 324. Card No. 8.	C., B. & Q. Mogul, No. 324. Card No. 59.	Baldwin No. 82. Card No. 8.
Diameter of high pressure cylinder	14 in.	20	20	14	14	20	20	14
Diameter of low pressure cylinder	24 in.	29	29	24	24	29	29	24
Stroke of cylinder	24 in.	24	26	24	24	24	24	24
Ratio of cylinder volumes	2.93	2.1	2.103	2.93	293	2.1	2.1	2.93
Volume of high pressure cylinder—cubic in.	3693	7550	8167	3694	3693	7550	7550	3693
Volume of low pressure cylinder—cubic in.	10857	15850	17170	10857	10857	15850	15850	10857
Volume of high pressure cylinder—cubic ft.	2.14	4.364	4.727	2.14	2.14	4.364	4.364	2.14
Volume of low pressure cylinder—cubic ft.	6.28	9.174	9.940	6.28	6.28	9.174	9.174	6.28
Length of high pressure indicator card on scale of 50	150	159	170	151	175(60)	158	158	152
Length of low pressure indicator card on scale of 50	150	156	170	151	175(60)	158	158	152
Per cent. of clearance high pressure cylinder	7.7	14.	11.2	7.7	7.7	14	14	7.7
Per cent. of clearance low pressure cylinder	6.5	9.7	5.9	6.5	6.5	9.7	9.7	6.5
Per. cent of clearance in valve based on h. p. cyl.	20	......	......	20	20	......	......	20
Clearance high pressure cylinder on scale of 50	11.5	22.6	19.0	11.6	13.5(60)	22.1	22.1	11.7
Clearance low pressure cylinder on scale of 50	9.82	15.1	10.0	9.82	11.5(60)	15.3	15.3	9.83
Clearance of valve on scale of 50	30	......	......	30.2	35(60)	......	......	30.4
Log. of volume of high pressure cylinder divided by length of high pressure indicator card on scale of 50.*	2.1538	2.4385	2.4442	2.1514	....	2.4412	2.4412	2.1486
Weight of steam after cut-off h. p. cylinder—lbs.	5913	.7349	1.1900	.5703	.4773	.8428	.9752	.4272
Weight of steam during compression h. p. cylinder	.0842	.2457	.2382	.0790	.0993	.2567	.2382	.0844
Weight of steam used per stroke h. p. cylinder	.5071	.4892	.9518	.4999	.3780	.5861	.7372	.3428
Weight of steam in l. p. cylinder after cut-off	.5740	.7404	1.0136	.5627	.5074	.8286	.8931	.4751
Log. of unit vol.* l. p.	2.6219	2.7695	2.7670	2.6190	....	2.7639	2.7639	2.6162
Speed in revolutions per minute	140.1	173.4	192	121.4	186.7	243.9	162.6	210.2
Weight of steam in l. p. cylinder during compression	.0832	.1403	.1557	.0601	.1022	.1655	.1406	.1101
Weight of steam used per stroke l. p. cylinder	.4908	.6001	.8579	.5026	.4052	.6631	.7525	.3650

	Baldwin No. 82. Card No. 48.	Schenectady, 12 wheeler. Cards 6 & 6a.	Schenectady, 12 wheeler. Cards 7 & 7a.	Schenectady, 12 wheeler. Cards 2 & 2a.	Schenectady, 10 wheeler. Card No. 61.	Schenectady, 10 wheeler. Card No. 74.	Schenectady, 10 wheeler. Card No. 80.
Diameter of high pressure cylinder	14	20	20	20	20	20	20
Diameter of low pressure cylinder	24	29	29	29	29	29	29
Stroke of cylinder	24	26	26	26	24	24	24
Ratio of cylinder volumes	2.93	2.103	2.103	2.103	2.1	2.1	2.1
Volume of high pressure cylinder—cubic in.	3693	8167	8167	8167	7550	7550	7550
Volume of low pressure cylinder—cubic in.	10857	17170	17170	17170	15850	15850	15850
Volume of high pressure cylinder—cubic ft.	2.14	4.727	4.727	4.727	4.364	4.364	4.364
Volume of low pressure cylinder—cubic ft.	6.28	9.940	9.940	9.940	9.174	9.174	9.174
Length of high pressure indicator card on scale of 50	151	170	170	169	163	163	163
Length of low pressure indicator card on scale of 50	151	170	170	168	163	163	163
Per cent. of clearance high pressure cylinder	7.7	11.2	11.2	11.2	9.9	9.9	9.9
Per cent. of clearance low pressure cylinder	6.5	5.9	5.9	5.9	4.7	4.7	4.7
Per. cent of clearance in valve based on h. p. cyl.	20	......	......	......	......	......	......
Clearance high pressure cylinder on scale of 50	11.6	19.0	19.0	18.9	16.14	16.14	16.14
Clearance low pressure cylinder on scale of 50	9.82	10.0	10.6	9.9	7.66	7.66	7.66
Clearance of valve on scale of 50	30.2	......	......	......	......	......	......
Log. of volume of high pressure cylinder divided by length of high pressure indicator card on scale of 50.*	2.1514	2.4442	2.4442	2.4466	2.4277	2.4277	2.4277
Weight of steam after cut-off h. p. cylinder—lbs.	.5804	1.3540	1.2390	.9839	1.0329	.9038	1.0470
Weight of steam during compression h. p. cylinder	.0804	.1966	.2043	.1817	.2409	.2010	.2223
Weight of steam used per stroke h. p. cylinder	.5000	1.1574	1.0347	.8022	.7920	.7028	.8247
Weight of steam in l. p. cylinder after cut-off	.5671	1.1693	1.1285	.8218	.8365	.7596	.8425
Log. of unit vol.* l. p.	2.6190	2.7670	2.7670	2.7720	2.7504	2.7504	2.7504
Speed in revolutions per minute	140.1	156	180	150	100	152	124
Weight of steam in l. p. cylinder during compression	.0740	.1074	.1291	.1115	.1221	.1233	.1166
Weight of steam used per stroke l. p. cylinder	.4931	1.0646	.9994	.7103	.7144	.6363	.7259

*The unit volume is the volume of the cylinder divided by the length of the indicator card on a scale of 50 parts to one inch.

TABLE N—*Continued.*

	Baldwin Eng. No. 82, on Erie tests, Card No. 1.	Baldwin Eng. No. 82. Erie tests, Card No. 2.	Baldwin Eng. No. 82. Erie tests, Card No. 3.	Baldwin Eng. No. 82. Erie tests, Card No. 4.	Baldwin Eng. No 82. Erie tests, Card No. 5.	Great Eastern Worsdell Comp. Eng. No. 230, Card No. 3.	Great Eastern Worsdell Comp. Eng. No. 230, Card No. 4.	Great Eastern Worsdell Comp. Eng. No. 230, Card No. 5.	Rhode Island Comp., on Brooklyn Elevated, Card No. 12.	Rhode Island Comp., on Brooklyn Elevated, Card No. 67.	Mexican Central, Johnstone Comp., Card No. 1.	Mexican Central, Johnstone Comp., Card No. 2.	Mexican Central, Johnstone Comp., Card No. 3.	Mexican Central, Johnstone Comp., Card No. 4.
Diameter of high pressure cylinder	14	14	14	14	14	18	18	18	11½	11½	13¾	13¾	13¾	13¾
Diameter of low pressure cylinder	24	24	24	24	24	26	26	26	18	18	29½	29½	29½	29½
Stroke of cylinder	24	24	24	24	24	24	24	24	16	16	24	24	24	24
Ratio of cylinder volumes	2.93	2.93	2.93	2.93	2.93	2.08	2.08	2.08	2.45	2.45	3.13	3.13	3.13	3.13
Volume of high pressure cylinder—cubic in	3693	3693	3693	3693	3693	6108.5	6108.5	6108.5	1660.8	1660.8	3552	3552	3552	3552
Volume of low pressure cylinder—cubic in	10857	10857	10857	10857	10857	12742.3	12742.3	12742.3	4071	4071	11124	11124	11124	11124
Volume of high pressure cylinder—cubic ft	2.14	2.14	2.14	2.14	2.14	3.535	3.535	3.535	9611	.9611	2.055	2.055	2.055	2.055
Volume of low pressure cylinder—cubic ft	6.28	6.28	6.28	6.28	6.28	7.374	7.374	7.374	2.356	2.356	6.437	6.437	6.437	6.437
Length of high pressure indicator card on scale of 50	155	154	154	155	154	165	157	161	154	152	130	130	130	130
Length of low pressure indicator card on scale of 50	155	154	154	155	154	165	157	161	154	152	130	130	130	130
Per cent. of clearance high pressure cylinder	7.7	7.7	7.7	7.7	7.7	5.88	5.88	5.88	11	11	6.3	6.3	6.3	6.3
Per cent. of clearance low pressure cylinder	6.5	6.5	6.5	6.5	6.5	4.76	4.76	4.76	10.2	10.2	18.9	18.9	18.9	18.9
Per cent. of clearance in valve based on h. p. cylinder.	20	20	20	20	20	......	......	......	......	......	......	......	......	......
Clearance high pressure cylinder on scale of 50	11.94	11.9	11.9	11.94	11.9	9.7	9.2	9.5	16.9	16.7	8.2	8.2	8.2	8.2
Clearance low pressure cylinder on scale of 50	10.08	10.0	10.0	10.08	10.0	7.8	7.5	7.6	15.7	15.5	24.6	24.6	24.6	24.6
Clearance of valve on scale of 50	31.00	30.8	30.8	31.00	30.8	......	......	......	......	......	......	......	......	......
Log. of volume of h. p. cylinder divided by length of h. p. indicator card on scale of 50	2.1401	2.1429	2.1429	1.1401	2.1429	2.3309	2.3525	2.3417	3.7953	3.8010	2.1987	2.1987	2.1987	2.1987
Weight of steam after cut-off h. p. cylinder—lbs	.5364	.5150	.5082	.4441	.4692	.5826	.5728	.4701	.1980	.2952	.6247	.6245	.6638	.6208
Weight of steam during compression h. p. cylinder	.0796	.0830	.0789	.0942	.1087	.0716	.0736	.0783	.0404	.0187	.0317	.0298	.0294	.0321
Weight of steam used per stroke h. p. cylinder	.4568	.4320	.4293	.3499	.3605	.5110	.4992	.3918	.1576	.2765	.5930	.5947	.6344	.5887
Weight of steam in l. p. cylinder after cut-off	.5203	.5324	.4871	.4499	.4619	.5027	.5142	.4203	.1799	.2810	.8000	.7647	.7969	.7445
Log. of unit vol. l. p.	2.6077	2.6105	2.6105	2.6077	2.6105	2.6502	2.6718	2.6609	2.1848	2.1905	2.6948	2.6948	2.6948	2.6948
Speed in revolutions per minute	120	160	160	140	172	252	192	264	306	180	72	60	57	66
Weight of steam in l. p. cylinder during compression	.0752	.0872	.0865	.0935	.0938	.0441	.0533	.0813	.0378	.0171	.1082	.1036	.0829	.1046
Weight of steam used per stroke l. p. cylinder	.4451	.4452	.4007	.3564	.3681	.4586	.4609	.3390	.1421	.2639	.6918	.6611	.7140	.6399

weight of steam shown to be in the cylinder at the termination of the compression period is subtracted from the weight of steam in the cylinders near the end of the expansion period. The remainder is taken as representing the amount of steam used in the cylinders, as shown by the indicator cards, 30–31, 42.

Table O shows the distribution of the steam at different parts of the stroke for an indicator card from the Vauclain engine. See Fig. 26.

TABLE O.

Giving Calculated Weight of Steam at Different Points of an Indicator Card from Vauclain Compound No. 82 in C. B. & Q. Tests.

Weight of steam at point 3, the terminal of expansion in h. p. cylinder..................................	.5913 lbs
Weight of steam in valve at H, the cut-off in l. p. cylinder	.0434 "
Weight of steam in l. p. clearance space at L........	.0832 "
Ratio of weight of steam in valve at H to the weight of steam discharged by the h. p. cylinder at point 3....	7.3%
Ratio of weight of steam in l. p. clearance space at L to the weight of steam discharged by the h. p. cylinder at point 3..................................	14.1%
Total addition to weight of steam discharged from h. p. cylinder at point 3 resulting from the admixture with the steam in valve and that in l. p. clearance space. Based on the measurement of the weight in valve at H and in the l. p. clearance at L, no allowance being made for condensation.........................	21.4%
Actual addition to weight of steam discharged from h. p. cylinder at point 3, based on measurement of indicator card at K..................................	12.0%
Difference between the actual addition of steam weight measured at K and the addition that would be found if the steam at H in valve, and the steam at L in l. p. clearance space, had been retained without condensation or leakage and had been added to that incoming from the h. p. cylinder, 21.4—12=.......	9.4%

It has been claimed that the steam pressure in the valve when the h. p. cylinder exhausts at point 3 is the same as the pressure of that exhaust, but in this case the valve pressure can be but 49 pounds absolute, or the same as that at H, while the pressure of the exhaust from the h. p. cylinder is that at point 3 or 126 pounds absolute.

The weight of the steam in the valve with 49 pounds pressure, absolute, is but .0434 pounds, while the weight with 126 pounds pressure would be .1205 pounds.

72. **Examples of Determination of Condensation, Leakage, and Re-Evaporation, from Various Indicator Cards.**—Table J gives some data about steam use in compound locomotives. The columns of particular interest are those which show the per cent. of increase or decrease of the steam used in the l. p. cylinder. It will be noticed that the cards taken from the two-cylinder compounds show less steam in the l. p. cylinder than in the h. p. This would point to a condensation somewhere, but it is not possible to say where it takes place without further analysis. It may be in the receiver, as in any compound engine with a receiver, if the receiver is not provided with an efficient re-heater, there is some loss of steam weight. This is very well shown in some tests of a triple expansion engine made by Professor Peabody, of the Massachusetts Institute. From the results of the analysis of the cards from the C., B. & Q. compound it will be noticed that the crack in the cylinder saddle caused so much leakage as to show a considerable increase of steam used in the l. p. cylinder. In closing upon this very important matter of the relative amounts of steam shown in the h. p. and l. p. cylinders, it is necessary to add that a 10 per cent. difference in the steam used by the two cylinders is not necessarily followed by a 10 per cent. loss of efficiency in the engine, and it may be that no material loss follows as much difference as this, for much depends upon the grade of expansion and conditions, 42, 44.

The object of making such analyses as this is, to learn about the rate of re-evaporation in compound locomotive cylinders. Re-evaporation must not be confused with leakage. With steam containing moisture when the volume increases the apparent weight of steam increases also. This arises from the fact that as the steam expands there is more heat in it than is necessary to keep the steam at the temperature corresponding to the reduced pressure, and also some heat is given back from the cylinder walls, which have been previously heated, and this extra heat goes to evaporate some of the moisture that is contained in the steam. This moisture results from the condensation while the steam is entering the cylinder from the boiler up to cut-off. The amount of steam condensed varies materially with different engines, but a rough approximation shows that the Baldwin engine on this test condensed something over 30 per cent. of the entering steam while the cylinders were being filled from *B* to *E*, Fig. 26. The condensation in some types of engine runs as high as 60 per cent., and in other engines, under particularly favorable conditions, as low as 20 per cent., and perhaps even lower in the first cylinder of the best designed triple expansion engines with steam jackets. Just as the steam condensed during admission evaporates during expansion, on account of the excess of heat over and above that necessary to keep the steam at a temperature corresponding to the pressure, and further by the heat received from the cylinder walls, so during compression some of the steam condenses by reason of the heat taken from it to heat up the cylinder head, the piston head, and the walls of the steam passage. These analyses of loss of heat, and the corresponding loss in steam weight, are interesting mainly in showing that the actual steam lines do not correspond with the usual theoretical steam line drawn for the sake of comparison on combined indicator cards, 43.

The hyperbola which is frequently drawn to show whether the engine leaks or not, does not take into account the full change in temperature during expansion, 41, 43. The adiabatic curve is an approximate curve which approaches very closely to the theoretical expansion of steam while doing work when there is no loss or gain of heat due to the heating of cylinder walls, etc. It takes account of the heat taken from the steam to do work. Owing to the re-evaporation in steam cylinders, it is generally the case that the hyperbola corresponds more nearly to the actual expansion line on an indicator card than does the adiabatic. In compression the actual compression line differs widely from both the hyperbola and the adiabatic, 6.

The weights of the steam present in the cylinders have been calculated for several points during the expansion of the steam in the two cylinders, Fig. 26, and are as follows:

The weight at point 1 is .57 pounds; at the point 2 it is .58 pounds; at the point 3 it is .59 pounds. Thus is shown the continual re-evaporation and corresponding increase in apparent steam weight during expansion in the h. p. cylinder.

The subject of cylinder condensation is a very complex one and cannot be treated here from a theoretical standpoint, as theoretical studies of the subject are of little value unless the constants of heat absorption are known. These have never been determined for locomotives. The most practical instruction is: insulate all exposed hot surfaces of the boiler and live steam passages and receptacles as fully as the best insulation will allow, and do this regardless of cost where fuel is high in price, 70.

CHAPTER XI.

THE VALVE GEAR ADJUSTMENTS.

It has been shown that when the valve motion is good and the receiver is of large volume, the division of the total work between h. p. and l. p. cylinders can be equalized with sufficient approximation for practical work by adjusting the cut-off in the cylinders. This is readily accomplished for locomotives that run always in the same direction by adjusting some part of the valve gear without increasing the complication. This is true of the Stephenson, Allen, Joy, Walscheart, and other positive motions. It is generally accomplished by changing the position of one of the links with respect to the other, either by shortening or lengthening the link hanger, or by off-setting one of the arms of the reverse shaft. Several modifications of the link motion that have been adopted to change the relative cut-off in the cylinders will be given in what follows.

For locomotives that run in both directions the adjustment of the cut-off is more difficult, and the devices for doing this introduce some new details of construction and are in some cases complicated. The simplest way in which to get a difference in cut-off in the cylinders, in both forward and back motion, for locomotives that run in both directions, is to give a different valve travel or outside lap to the different cylinders. In all cases not enough difference can be produced in this way to accomplish the desired result without making the steam distribution in one cylinder much less efficient than in the other, but where the cylinder ratio is selected within the proper limits, and the receiver has sufficient capacity, and the valve travel and steam ports are ample, the adjustment can be made with perfect satisfaction by changing the travel or outside lap to adjust the cut-off, 45–56, 77–81. In Tables P, Q, R, S, T, U, U1 and V, will be found the result of some changes of this kind and the opinions of various designers on this matter. With the Joy gear, the variation in cut-off may be produced by inclining the sliding links to each other.

73. **Mallet's System of Cut-Off Adjustment.**—In the earlier Mallet engines the lifting shaft is divided so that the valve motion of each cylinder is to a certain extent

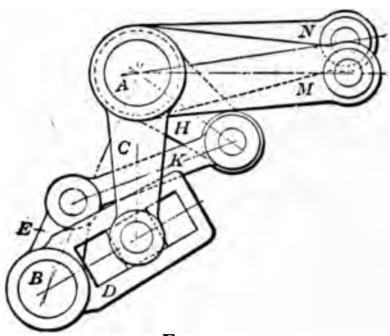


FIG. 34.
Mallet Regulating Device.

independent of the other. The h. p. valve gear is controlled by a screw and nut, which takes the place of the ordinary quadrant. The nut which is on the h. p. reverse lever carries a short sector or quadrant, and a latch on the l. p. reverse lever works in this sector. The effect is that both cylinders can be reversed by moving the h. p. lever; while by adjusting the l. p. lever the cut-off in that cylinder may be made either later or earlier than in the h. p. cylinder.

Mr. Mallet has adopted a differential motion for the purpose of obtaining a later cut-off in the l. p. cylinder in both forward and backward gear. The principle of this motion is illustrated by Fig. 34. In this Fig., *A* is the lifting shaft and *B* is an auxiliary shaft. The lifting arm *M* of the h. p. link and the arm *C* are keyed to the lifting shaft, while the l. p. lifting arm *N* and the arm *H* are in one piece, which turns about this shaft. The slotted arm *D* and the arm *E* are keyed to the auxiliary shaft. The arm *C* carries a block which slides in the slotted piece *D*. The parts are shown in Fig. 34 in a position for backing, the l. p. link being raised higher than the h. p. link and therefore cutting off later. In full backing gear the arms *M* and *N* would be parallel and hence give the same cut-off in both cylinders. In mid-gear the arms *C* and *D* are on the center line *A B*, while in forward gear or to the left, the lifting arm *N* is lowered more rapidly than the arm *M*. Mr. Mallet gives the following as the distribution obtained with this arrangement:

	Forward Gear.						Backward Gear.		
High-pressure cylinder......	.70	.60	.50	.40	.30	.0	.0	.60	.70
Low-pressure cylinder.......	.70	.65	.60	.55	.50	.0	.0	.65	.70

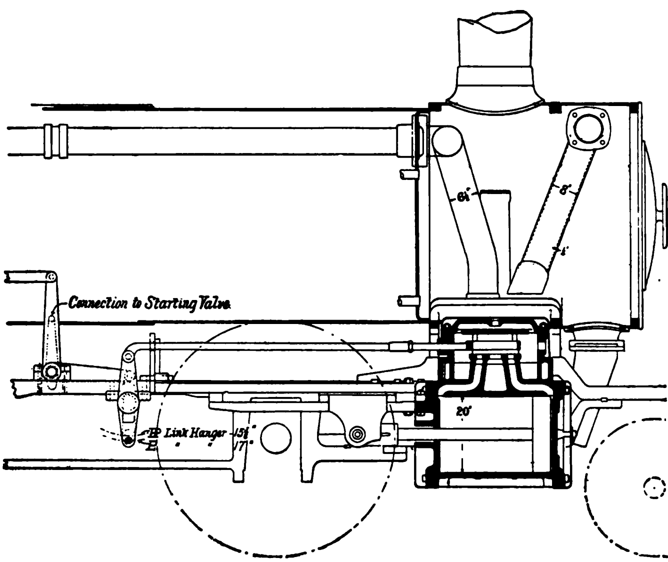


FIG. 35.
C. B. & Q. Link Hanger Adjustment.

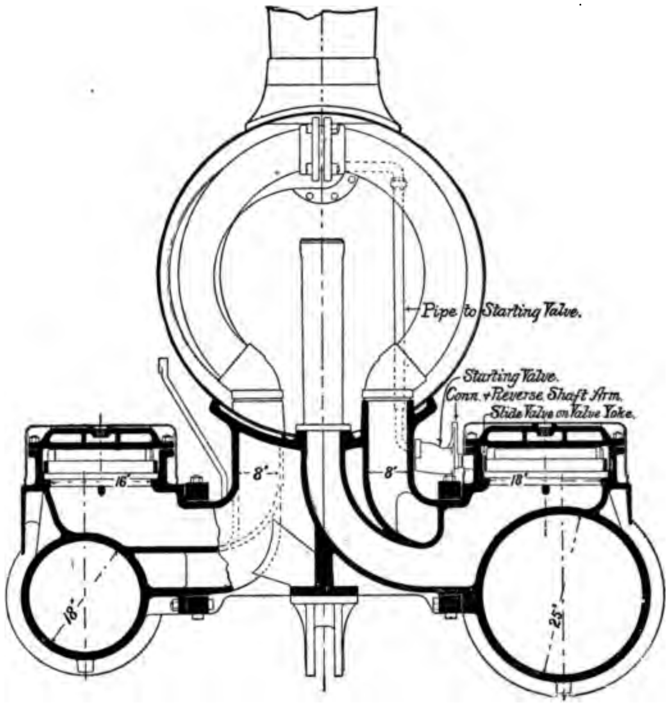


FIG. 36.
Chicago, Burlington & Quincy Gear.

74. **Chicago, Burlington & Quincy System.**—Mr. William Forsyth, Mechanical Engineer of the Chicago, Burlington & Quincy Railroad, has designed a variable cut-off gear for the two cylinders of a Lindner compound by making one of the reverse shaft arms loose on the shaft. The loose arm is a bell crank with a vertical arm similar to the one used for reverse shafts on American engines. From the top of the loose arm a short reach rod runs back about four feet, and is there attached to the main reach rod running to the reverse lever. With this arrangement, by making one of the vertical arms shorter than the other, a movement of the reverse lever causes a different angle of rotation of the two

reverse shaft arms, and one link can be dropped lower than the other while running in either direction. This arrangement worked satisfactorily and the distribution was excellent. It was found, however, that the lengthening of the l. p. link hanger accomplished the same end, for regular freight engines, and the second compound was built with the hangers at different length, as given in Figs. 35 and 36.

75. **Heintzelman System.**—On the Southern Pacific the following plan for adjusting the cut-off has been devised by Mr. T. W. Heintzelman. See Figs. 37 and 38.

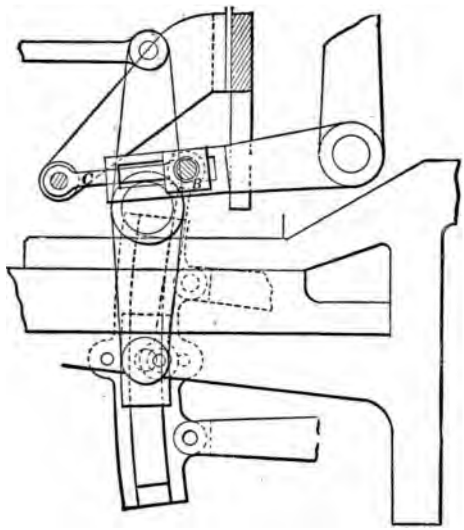


FIG. 37.
Heintzelman Gear.

The horizontal arm of the reverse shaft has a slot in which slides a block. To this block is attached the upper end of the link hanger, and also one end of a horizontal link. The horizontal link at the other end is attached to a bracket on the guide yoke or any other convenient part of the locomotive. This device is put on the h. p. side of the engine. Referring to Figs. 37 and 38 it will be seen that the link block is shown in the centre of the link. It is evident that if the reverse shaft be dropped from the position shown, the block in the slot in the reverse shaft arm, as well as the upper end of the link hanger, will be pulled, by means of the horizontal link attached to the bracket, to a

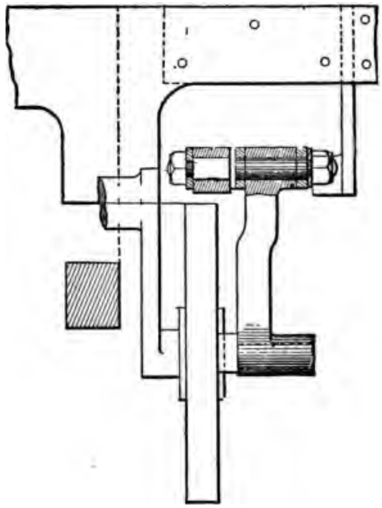


FIG. 38.
Heintzelman Gear.

position further to the left, or toward the end of the reverse shaft arm, than is shown. Meantime, the upper end of the link hanger on the other side of the engine has remained at the same distance from the centre of the reverse shaft. The effect of dropping the horizontal reverse shaft arm to the lowest position to put the engine in full forward gear, is to bring the upper ends of both link hangers in the same relative position with respect to the reverse shaft, and give the same cut-off in both cylinders in full forward gear. At all other positions of the link, the block in the reverse shaft on the h. p. side is nearer the centre of the reverse shaft, and the effect is the same as if a shorter reverse shaft arm, and one of variable length, was used on the h. p. side. In this way the l. p. link is lower than the h. p. link, for all cut-offs except that of full forward gear, hence the cut-off is longer in the l. p. cylinder than in the h. p. The effect of this device on the distribution of steam power in the cylinders and on the relative cut-offs is given in Table P.

TABLE P.

Heintzelman djustment of Cut-off and Per Cent. of Power in H. P. and L. P. Cylinders. See Appendix R.

Cut-off h. p. cylinder, inches.	Cut-off l. p. cylinder, inches.	Per cent. of total work done in h. p. cylinder.	Per cent. of total work done in l. p. cylinder.
23⅜	23⅜	41.15	58.85
22½	22⅞	43.18	56.82
20⅝	21⅞	41.64	58.36
15	18½	46.28	53.72
12½	17	45.04	54.96
9⅞	15	49.08	50.92

76. **The Rogers Locomotive Works Link Hanger Adjustment.**—The Rogers Locomotive Works have used a link hanger in two parts, each part being provided with teeth to prevent slipping. In this way the bolts can be loosened and the link hanger be made longer or shorter as desired.

76a. **Different Adjustments of Cut-Offs That Have Been Used for Compound Locomotives.**—Mr. von Borries from his experience has finally settled on the following ratio of cut-offs in the h. p. and l. p. cylinders as being in his opinion best adapted for average work. See Fig. 27.

Cut-off H. P. Cylinder, per cent.	30	40	50	60	70	78
" L. P. " "	40	50	58	65	73	80

After a number of experiments Mr. Joseph Lythgoe, of the Rhode Island Locomotive Works has decided to use 1¼ in. outside lap on the h. p. cylinder, and ⅞ in. lap on l. p. cylinder. This gives about 3¾ in. later cut-off in the l. p. cylinder for a 24 in. stroke, and it is believed will so satisfactorily adjust the cut-offs that a change in the length of the link hanger will not be needed. This plan has the advantage of giving the same relative cut-offs in both cylinders whether the engine is going ahead or backing. The valve travel used with this amount of outside lap is 6¼ ins.

TABLE Q.

Giving details of Valve Movement and Port Openings on Dean Compound. Locomotive on Old Colony R. R. Cylinders 20 in. and 28 in. × 24 in. Drivers 69 in. Diameter. Valve Travel $6\frac{1}{4}$ in. Outside Lap 1 in. Inside Clearance or Negative Lap $\frac{1}{4}$ in. See Appendix R.

	h. p Cut-off	l. p. Cut-off	h. p. Lead.	l. p. Lead.	h. p. Release.	l. p. Release.	h. p. Compression.	l. p. Compression.	Port Opening, h. p.	Port Opening. l. p.
Forward End.	$20\frac{7}{8}$	$21\frac{3}{8}$	$\frac{1}{64}$	$\frac{1}{64}$	$22\frac{5}{8}$	$22\frac{7}{8}$	$23\frac{3}{16}$	$23\frac{9}{16}$	$1\frac{7}{8}$	$2\frac{1}{8}$
	18	$19\frac{1}{16}$	$\frac{3}{32}$	$\frac{3}{32}$	$21\frac{1}{8}$	$21\frac{3}{4}$	$22\frac{3}{4}$	23	$1\frac{1}{8}$	$1\frac{5}{16}$
	16	$17\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{8}$	$20\frac{1}{8}$	$20\frac{7}{8}$	$22\frac{1}{4}$	$22\frac{5}{8}$	$\frac{13}{16}$	1
	$13\frac{7}{8}$	$15\frac{7}{8}$	$\frac{5}{32}$	$\frac{5}{32}$	19	20	$21\frac{9}{16}$	$22\frac{1}{8}$	$\frac{5}{8}$	$\frac{13}{16}$
	$11\frac{7}{8}$	$14\frac{1}{8}$	$\frac{5}{32}$	$\frac{5}{32}$	$17\frac{7}{8}$	$19\frac{1}{4}$	$20\frac{15}{16}$	$21\frac{11}{16}$	$\frac{1}{2}$	$\frac{21}{32}$
	$9\frac{9}{16}$	$12\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$16\frac{9}{16}$	$18\frac{1}{4}$	$20\frac{1}{16}$	$21\frac{1}{8}$	$\frac{13}{32}$	$\frac{1}{2}$
	$7\frac{5}{8}$	$10\frac{5}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$15\frac{1}{4}$	$17\frac{3}{16}$	$19\frac{1}{4}$	$20\frac{3}{8}$	$\frac{11}{32}$	$\frac{7}{16}$
Back End.	$20\frac{5}{8}$	$21\frac{1}{8}$	$\frac{1}{64}$	$\frac{1}{64}$	$22\frac{1}{2}$	$22\frac{5}{8}$	$23\frac{3}{8}$	$23\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{3}{16}$
	$17\frac{7}{8}$	$18\frac{7}{8}$	$\frac{3}{32}$	$\frac{3}{32}$	21	$21\frac{11}{16}$	$22\frac{11}{16}$	$22\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{5}{16}$
	$16\frac{1}{16}$	$17\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{8}$	$20\frac{1}{16}$	$20\frac{15}{16}$	$22\frac{3}{16}$	$22\frac{1}{2}$	$\frac{13}{16}$	1
	$14\frac{1}{16}$	$15\frac{7}{8}$	$\frac{5}{32}$	$\frac{5}{32}$	$19\frac{1}{16}$	$19\frac{3}{4}$	$21\frac{9}{16}$	$22\frac{1}{8}$	$\frac{5}{8}$	$\frac{27}{32}$
	12	$14\frac{5}{16}$	$\frac{5}{32}$	$\frac{5}{32}$	$17\frac{7}{8}$	$19\frac{1}{4}$	$20\frac{15}{16}$	$21\frac{11}{16}$	$\frac{1}{2}$	$\frac{21}{32}$
	$9\frac{15}{16}$	$12\frac{3}{4}$	$\frac{3}{16}$	$\frac{3}{16}$	$16\frac{11}{16}$	$18\frac{3}{8}$	$20\frac{1}{4}$	$21\frac{1}{4}$	$\frac{13}{32}$	$\frac{17}{32}$
	$7\frac{9}{16}$	$10\frac{7}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$15\frac{3}{8}$	$17\frac{1}{2}$	$19\frac{7}{16}$	$20\frac{1}{2}$	$\frac{11}{32}$	$\frac{7}{16}$

TABLE R.

Giving Details of Valve Movement and Port Openings on Dean Compound Locomotive on Lehigh Valley R. R. Cylinders 20 in. and 30 in. × 24 in. Drivers 50 in. Diameter. Valve Travel h. p. 5 in., l. p. $6\frac{1}{4}$ in. Outside Lap h. p. $\frac{3}{4}$ in., l. p. $1\frac{1}{8}$ in. Inside Clearance or Negative Lap h. p. $\frac{3}{16}$ in., l. p. 0 in. See Appendix R.

	h. p. Cut-off	l. p. Cut-off	h. p. Lead.	l. p. Lead.	h. p. Release.	l. p. Release.	h. p. Compression.	l. p. Compression.	h. p. Port Opening.	l. p. Port Opening.
Forward End.	20	$19\frac{3}{8}$	$\frac{1}{64}$	Line	$22\frac{1}{2}$	$22\frac{1}{2}$	23	$22\frac{9}{16}$	$1\frac{13}{16}$	$1\frac{17}{32}$
	18	$16\frac{9}{16}$	$\frac{9}{64}$	$\frac{1}{8}$	$21\frac{5}{8}$	$21\frac{7}{16}$	$22\frac{3}{8}$	$21\frac{7}{16}$	$1\frac{7}{32}$	$1\frac{1}{32}$
	16	$13\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$20\frac{5}{8}$	$20\frac{1}{2}$	$21\frac{11}{16}$	$20\frac{3}{8}$	$1\frac{5}{16}$	$\frac{3}{4}$
	14	11	$\frac{15}{64}$	$\frac{1}{4}$	$19\frac{5}{8}$	19	$20\frac{7}{8}$	$19\frac{1}{8}$	$\frac{3}{4}$	$\frac{9}{16}$
	12	$8\frac{1}{2}$	$\frac{1}{4}$	$\frac{9}{32}$	$18\frac{5}{8}$	$17\frac{3}{4}$	20	$17\frac{13}{16}$	$\frac{5}{8}$	$\frac{15}{32}$
	$9\frac{7}{8}$	$6\frac{1}{4}$	$\frac{9}{32}$	$\frac{19}{64}$	$17\frac{1}{2}$	$16\frac{5}{16}$	19	$16\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{8}$
Back End.	$19\frac{3}{4}$	$19\frac{7}{8}$	$\frac{1}{64}$	Line	$22\frac{1}{2}$	$22\frac{13}{16}$	$22\frac{13}{16}$	$22\frac{3}{4}$	$1\frac{5}{16}$	$1\frac{3}{4}$
	17	$18\frac{5}{8}$	$\frac{3}{32}$	$\frac{1}{8}$	$21\frac{7}{8}$	$22\frac{3}{8}$	$22\frac{1}{4}$	$22\frac{5}{16}$	$1\frac{1}{16}$	$1\frac{7}{16}$
	$15\frac{3}{4}$	$17\frac{5}{8}$	$\frac{1}{8}$	$\frac{5}{32}$	$21\frac{3}{8}$	$21\frac{15}{16}$	$21\frac{3}{4}$	$21\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$
	14	$16\frac{1}{2}$	$\frac{11}{64}$	$\frac{3}{16}$	$20\frac{3}{4}$	$21\frac{1}{2}$	$21\frac{1}{4}$	$21\frac{1}{2}$	$\frac{3}{4}$	$1\frac{3}{32}$
	12	$14\frac{15}{16}$	$\frac{13}{64}$	$\frac{7}{32}$	$19\frac{13}{16}$	$20\frac{7}{8}$	$20\frac{3}{4}$	$20\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{8}$
	$9\frac{7}{8}$	$13\frac{5}{16}$	$\frac{15}{64}$	$\frac{1}{4}$	$18\frac{3}{4}$	$20\frac{3}{16}$	$19\frac{7}{16}$	20	$\frac{1}{2}$	$\frac{3}{4}$

TABLE S.

Giving the Steam Port Openings of Schenectady (Pitkin) Compouna Locomotive on Chicago & North-Western R. R. Cylinders 20 in. and 30 in. × 24 in. Drivers 68 in. Diameter. Valve Travel 6½ in. Outside Lap 1⅛ in. Inside Clearance or Negative Lap, h. p. ⅛ in., l. p. 1/16 in.

h. p. cyl. Cut-off Inches.		l. p. cyl. Cut-off Inches.		h. p. cyl. Lead, Inches.		l. p. cyl. Lead, Inches.		h. p cyl. Valve Opening, Inches.		l. p. cyl. Valve Opening, Inches.	
Front Stroke.	Back Stroke.	Front Stroke.	Back Stroke.	Front Stroke.	Back Stroke.	Front Stroke.	Back Stroke.	Front Stroke.	Back Stroke.	Front Stroke.	Back Stroke.
19 15/16"	20 3/16"	20 9/16"	20 13/16"	1/32"	1/32"	1/32"	1/32"	1 13/16"	1 13/16"	1 5/32"	1 3/32"
19¼"	19 7/16"	20"	20¼"	3/32"	3/32"	1/16"	1/16"	1 9/16"	1 17/32"	1⅞"	1 13/16"
16⅝"	16¾"	18"	18¼"	⅛"	⅛"	⅛"	⅛"	1"	31/32"	1¼"	1¼"
14 1/16"	14⅛"	16"	16⅛"	5/32"	5/32"	⅛"	⅛"	11/16"	11/16"	29/32"	⅞"
11 9/16"	11 9/16"	14"	14"	5/32"	5/32"	5/32"	5/32"	½"	½"	11/16"	11/16"
9⅜"	9¼"	12	12	3/16"	3/16"	5/32"	5/32"	⅜"	⅜"	17/32"	17/32"

TABLE T.

Giving the Steam Port Openings of the Schenectady Compound Locomotive on Adirondack & St. Lawrence R. R. Cylinders 19 in. and 28 in. × 24 in. Drivers, 69 in. diameter. Valve Travel, 6½ in. Outside Lap, 1⅛ in. Inside Clearance or Negative Lap h. p., ⅛ in.; l. p., 1/16 in. See Appendix R.

h. p. cyl. Cut-off, Inches.		l. p. cyl. Cut-off, Inches.		h. p. cyl. Lead, Inches.		l. p. cyl. Lead, Inches.		h. p. cyl. Valve Opening, Inches.		l. p. cyl. Valve Opening, Inches.	
Front Stroke	Back Stroke	Front Stroke	Back Stroke	Front Stroke	Back Stroke	Front Stroke	Back Stroke	Front Stroke.	Back Stroke	Front Stroke.	Back Stroke.
20 1/16"	20 5/16"	20½"	20 13/16"	1/32"	1/32"	1/32"	1/32"	1⅞"	1 13/16"	2⅛"	2⅛"
19 5/16"	19 11/16"	20"	20⅜"	1/16"	1/16"	1/16"	1/16"	1 11/16"	1⅝"	1⅞"	1⅞"
17 1/16"	17 3/16"	18"	18¼"	⅛"	⅛"	⅛"	⅛"	1 3/32"	1 1/32"	1 5/16"	1 5/16"
14 13/16"	14 11/16"	16"	16¼"	5/32"	5/32"	5/32"	5/32"	25/32"	23/32"	15/16"	15/16"
12 5/16"	12"	14"	14⅛"	3/16"	3/16"	3/16"	3/16"	19/32"	17/32"	23/32"	23/32"
9 11/16"	9 5/16"	12"	12 1/16"	3/16"	3/16"	3/16"	3/16"	17/32"	15/32"	9/16"	9/16"

TABLE U.

Giving the Steam Port Openings of Schenectady Compound Locomotive on Adirondack & St. Lawrence R. R. Cylinders, 22 in. and 32 in. × 26 in. Drivers, 51 in. diameter. Valve travel, 5½ in. Outside Lap, ¾ in. Inside Clearance or Negative Lap, 0 in. See Appendix R.

h. p. cyl. Cut-off, Inches.		l. p. cyl. Cut-off, Inches.		h. p. cyl. Lead, Inches.		l. p. cyl. Lead, Inches.		h. p. cyl. Valve Opening, Inches.		l. p. cyl. Valve Opening, Inches.	
Front Stroke	Back Stroke	Front Stroke	Back Stroke	Front Stroke	Back Stroke	Front Stroke	Back Stroke	Front Stroke.	Back Stroke.	Front Stroke	Back Stroke.
23 5/16"	23⅝"	23¾"	23 15/16"	1/16"	1/16"	1/16"	1/16"	1¾"	1 11/16"	2"	2"
20¼"	20½"	21"	21⅜"	⅛"	⅛"	⅛"	⅛"	31/32"	15/16"	1 3/16"	1 3/16"
17½"	17¾"	19"	19 5/16"	5/32"	5/32"	5/32"	5/32"	21/32"	⅝"	⅞"	⅞"
14 13/16"	14 13/16"	17"	17 3/16"	3/16"	3/16"	3/16"	3/16"	15/32"	7/16"	11/16"	11/16"
12½"	12¼"	15"	15⅛"	3/16"	3/16"	3/16"	3/16"	⅜"	⅜"	9/16"	9/16"
10"	9½"	13"	13"	7/32"	7/32"	3/16"	3/16"	5/16"	5/16"	7/16"	7/16"

TABLE U1.

Giving the Cut-offs in High-Pressure and Low-Pressure Cylinders of Various Compounds; also, Ratio of Receiver Capacity to High-Pressure Cylinder. The Cut-offs are those measured from the Valves when the Engines are Cold and not Running. See Appendix R.

	Diam. of h. p. cyl. Inches.	Diam. of l. p. cyl. Inches.	Stroke of Piston. Inches.	Cut-off h. p. cyl. Inches.	Cut-off h. p. cyl. Per cent.	Cut-off l. p. cyl. Inches.	Cut-off l. p. cyl. Per cent.	Valve travel h. p. Inches.	Valve travel l. p. Inches.	Outside lap h. p. Inches.	Outside lap l. p. Inches.	Inside lap h. p. Inches.	Inside lap l. p. Inches.	Inside clearance h. p. Inches.	Inside clearance l. p. Inches.	Length of port h. p. Inches.	Length of port l. p. Inches.	Maker's estimate of equal single expansion cyls. Ins.	Kind of service. Passenger or Freight	Diameter of drivers.	Loco. Builder's opinion of most economical cut-off for Comps. h.p.cyl.	Loco. Builder's standard cut-off for Comps. l. p.cyl.	Loco. Builder's standard ratio of receiver capacity to capacity of h. p. cyl.
von Borries 2-cylinder Mogul for Jura-Berne-Lucerne Ry.	17.7	25.2	25.6		13		20												Frgt.	59.8			
					20		29																
					31		42																
					41		51																
					51		59																
					60		66																
					76		80																
Worsdell 2 cylinder for London & Northeastern Ry., England.	18	26	24		73		86							¼	⅛				Frgt.	56			
					50		73																
Worsdell 2 cylinder for London & Northeastern Ry., England.	18	26	24		70		84.5							¼	⅛				Pass.				
					50		73.																
Worsdell 2 cylinder for London & Northeastern Ry., England.	20	28	24		63		78							3/16	¼				Pass.	91¼			
					50		68																
					40		62.5																
					47		67.																
					53		70.																

Worsdell 2 cylinder for the Santa Fé & Cordoba Great Southern Ry.	16	23	24		73¾		78	4.19	4.50	1.12	1.12			¼		12.87	17					
					72¼		77½															
					61¼		67¼	3.50	3.72													
					58½		66															
					52¼		60¼	3.19	3.47													
					49		57¾															
					41½		51½	2.97	3.22													
					38½		48½															
					29½		40	2 72	2.97													
					27½		37¾															
					16½		27¾															
					18		27½	3.62	2.78													
Schenectady 10 wheel. Pitkin 2 cyl. for Pennsylvania R. R.	20	30	24					6½	6½	1⅛	1⅛			⅛	1-16	19	21		Pass.	74		
Schenectady 10 wheel. Pitkin 2 cylinder for Chicago & Northwestern Ry.	20	30	24	9 5-16	38¾	12	50	6½	6½	1⅛	1⅛			⅛	1-16	19	21	19	Pass.	68	4-10	5-10
Schenectady 10 wheel. Pitkin 2 cyl. for Adirondack & St. Lawrence R. R.	19	28	24	9½	39 3-5	12	50 1-5	6½	6½	1⅛	1⅛			⅛	1-16	18	20	18	Pass.	69	4-10	5-10
Schenectady Consolidation Pitkin 2 cyl. for Adirondack & St. Lawrence R. R	22	32	26	9¾	37	13	50	5½	5½	¾	¾					19	21	21	Frgt.	51	4-10	5-10
Schenectady Mogul. Pitkin 2 cyl. for Adirondack & St. Lawrence R. R.	20	30	26	21⅞		22¼		6½	6½	1⅛	1⅛			5-16	3-16	18	20		Frgt.	57		
				17		18⅛																
				13½		15⅛																
				12		13⅛																
				10¾		12⅜																
Schenectady 10 wheel, Pitkin 2 cyl. for Michigan Central R. R.	19	29	24	21⅞		22¼		6½	6½	1⅛	1⅛			5-16	3-16	18	20	18	Pass.		4-10	5-10
				17		18⅛																
				13½		15⅛																
				12		13⅛																
				10¾		12⅜																

TABLE U1—*Continued.*

	Diam. of h. p. cyl. Inches.	Diam. of l. p. cyl. Inches.	Stroke of Piston. Inches.	Cut-off h. p. cyl. Inches.	Cut-off h. p. cyl. Per cent.	Cut-off l. p. cyl. Inches.	Cut-off l. p. cyl. Per cent.	Valve travel h. p. Inches.	Valve travel l. p. Inches.	Outside lap h. p. Inches.	Outside lap l. p. Inches.	Inside lap h. p. Inches.	Inside lap l. p. Inches.	Inside clearance h. p. Inches.	Inside clearance l. p. Inches.	Length of port h. p. Inches.	Length of port l. p. Inches.	Maker's estimate of equal single expansion cyls. Ins.	Kind of service. Passenger or Freight.	Diameter of drivers.	Loco. Builder's standard cut-off for comps., h. p. cyl.	Loco. Builder's standard cut-off for comps. l. p. cyl.	Loco. Builder's standard ratio of receiver capacity to capacity of h. p. cyl.
Schenectady 12 wheel. Pitkin 2 cyl. for Southern Pacific R. R.	20	29	26	22¼		22⅝				⅞	⅞			¼	3/16				Frgt.	51			
				20¼		21																	
				17⅞		19																	
				15⅝		17																	
				15⅝		13																	
Schenectady 10 wheel. Pitkin 2 cyl. for Southern Pacific R. R.	20	29	24	20¼		20⅝				1⅛	1⅛			5-16	3-16	18	20		Pass.	69			
				19½		19⅞																	
				17½		18⅜																	
				15¼		16½																	
				12½		14⅛																	
				10¼		12⅜																	
Schenectady Consolidation, Pitkin 2 cyl. for Southern Pacific R. R. With Heintzelman's Valve Gear Adjustment.	20	29	24	23⅜		23⅜				¾	¾			⅜	3-16	18	20		Frgt.	51			
				22½		22⅞																	
				20⅝		21⅞																	
				15		18½																	
				12½		17																	
				9⅞		15																	
Schenectady 12 wheel. Pitkin 2 cyl. for Southern Pacific R. R.	20	29	26	Same in both Cylinders.						¾	¾			⅜	3-16	18	20		Frgt.	51			
Rhode Island 10 wheel. Batchellor 2 cyl. for C., M. & St. P. R. R.	21	31	26	11½		12¾		6¼	6¼	1	⅞			⅜	⅜	20	25	20	Pass.	78			Varies from 2 to 1.

Rhode Island 8 wheel. Batchellor 2 cyl. for N. Y., N. H. & Hartford R. R.	21	31	26	11½	......	12¾	......	6¼	6¼	1	⅞	....	..	⅜	⅜	20	25	......	Pass.	78	......	......	Varies from 2 to 1.
Rhode Island, Forney. Batchellor 2 cyl. for Union Elevated, Brooklyn.	11½	18	16	......	39 46 54.5 64 72 78 82.5 86	....	36.5 44 5 51 62 70 76.5 81 85	5	5	⅞	1	....	....	⅝	0	10	17	. ...	Pass.	42	......	......	
Chicago, Burlington & Quincy Mogul. Lindner 2 cyl. for C., B. & Q. R. R.	20	29	24	10 15½ 19¼	41 64 80	15 18¾ 21	62 78 87	4⅝	6	1 1/16	15/16	....	....	7-32	⅛	18	20	19×24	Frgt.	62	12	15	2.25
Pennsylvania R. R. 8 wheel Lindner 2 cyl. for Pa. R. R.	19½	31	28	22	78.6	23.5	83.9	7	7	1½	1 3/16	....	....	3-5	¾	29	29	......	Pass.	84	......	......	
Pittsburgh Mogul. Colvin 2 cyl. for —— R. R.	19	29	26	23½	90.4	24⅛	92.8	5	6	¾	¾	....	....	5-16	⅛	18	20	19×26	Frgt.	54	......	....	2.5
Pittsburgh 8 wheel American. Colvin 2 cyl.	19	29	26	21¾	83.7	23½	90 4	5	6	1	⅞	....	..	5-16	⅛	16	18	19×26	Pass.	72	......	....	2.5
Richmond 10 wheel. Mellin 2 cyl. for Chesapeake & O. R. R.	19	29	24	7 21	29 87	7 21	29 87	5½	5½	⅞	⅞	..	..	¼	0	18	20	19×24	Frgt.	57	......	....	Varies form 2 to 3.
Richmond 10 wheel. Mellin 2 cyl. for C. C. C. & St. L. R. R.	19	30	24	7 21	29 87	7 21	29 87	5½	5¾	⅞	⅞	..	....	5-16	0	23	23	19×24	Frgt.	56	......	......	Varies form 2 to 3.
Old Colony American. Dean 2 cyl. for Old Colony R. R.	20	28	24	10	41	12	50	6¼	6¼	1	1	....	..	¼	¼	20	24	18×24	P & F	69	33⅝	....	Varies form 3 to 4.

TABLE U1—*Continued.*

	Diam. of h. p. cyl. Inches.	Diam. of l. p. cyl. Inches.	Stroke of Piston. Inches.	Cut-off h. p. cyl. Inches.	Cut-off h. p. cyl. Per cent.	Cut-off l. p. cyl. Inches.	Cut-off l. p. cyl. Per cent.	Valve travel h. p. Inches.	Valve travel l. p. Inches.	Outside lap h. p. Inches.	Outside lap l. p. Inches.	Inside lap h. p. Inches.	Inside lap l. p. Inches.	Inside clearance h. p. Inches.	Inside clearance l. p. Inches.	Length of port h. p. Inches.	Length of port l. p. Inches.	Maker's estimate of equal single expansion cyls. Ins.	Kind of service. Passenger or Freight.	Diameter of drivers.	Loco. Builders standard cut-off for comps., h. p. cyl.	Loco. Builder's standard cut-off for comps., l. p. cyl.	Loco. Builder's standard ratio of receiver capacity to capacity of h. p. cyl.
Old Colony American. Dean 2 cyl. for Old Colony R. R.	20½	29	26	10	38½	13	50	6¼	6¼	1⅛	1⅛	....	....	3-16	0	20	24	18×26	Pass.	79	33⅓	......	Varies from 3 to 4.
Lehigh Valley Consolidated. Dean 2 cyl. for L. V. R. R.	20	30	24	14	58½	11	46	5	6¼	¾	1⅛	....	....	3-16	0	16	24	20×24	Frtg.	50	33⅓	......	Varies from 3 to 4.
Mexican Cent. R. R. Consolidation. Johnstone 4 cyl. for Mexican Central (No. 66 altered from single Expansion Loco.)	14	30¾	24	9	......	16¼													Frtg.	50		......	No receiver.
				10		17½																	
				12		18½																	
				14		20																	
				16		21																	
				18		21½																	
Rhode Island 10 wheel. Johnstone 4 cyl. for Mexican Central R. R.	13¾	29¾	24	19¾	......	22¾	......	5½	4½	1⅛	½	1-32	1-32			14	14	......	Frtg.	56	......	......	No receiver.
				18⅞		22¼																	
				18⅛		22¼																	
				17¼		22																	
				16		21⅝																	
				14¾		21⅛																	
				13⅛		20¼																	
				11½		19¾																	
				9¼		19																	
				6½		17⅞																	

Baldwin 10 wheel. Vauclain 4 cyl. for Chicago, Milwaukee & St. Paul R. R.	12	20	26	12.3		15		4½	4½	⅞	⅝								Frtgt.	62			No receiver.
				13.3		15.9																	
				14.3		16.9																	
				15.4		17.7																	
				16.		18.6																	
				17.4		19.2																	
				18.3		19.9																	
				18.9		20.4																	
				19.6		20.9																	
				20.7		21.8																	
				21.6		22.8																	
				22.1		23.2																	
Baldwin 8 wheel. Vauclain 4 cyl. for Central R. R. of New Jersey.	13	22	24	20 13/16		21 9/16		5¼	5¼	⅞	⅝	0	0			24	24	18½ × 24	Pass.	78			0
				20		20⅞																	
				19		20																	
				18		19⅛																	
				17		18½																	
				16		17⅝																	
				15		16¾																	No receiver.
				14		16																	
				13		15¼																	
				12		14⅜																	
				11		13½																	
				10		12⅝																	
				9		11¾																	
				8		10 9/16																	
				7		9⅝																	
				6		8 9/16																	
				5		7½																	
				4		6 7/16																	
Baldwin 6 wheel. Vauclain 4 cyl. for Ramal Dumont R. R., Brazil.	7½	13	18	15⅞		16 1/16		3⅝	3⅝	9-16	½	0	0			13	13		Frgt.	36			
				15		15⅜																	0
				14		14 7/16																	
				13		13½																	
				12		12 9/16																	No receiver.
				11		11⅝																	
				10		10 11/16																	
				9		9¾																	
				8		8¾																	
				7		7⅞																	
				6		6⅞																	
				5		5¾																	
				4		4¾																	
				3		3¾																	

TABLE U1—*Continued.*

	Diam. of h. p. cyl. Inches.	Diam. of l. p. cyl. Inches.	Stroke of Piston. Inches.	Cut-off h. p. cyl. Inches.	Cut-off h. p. cyl. Per cent.	Cut-off l. p. cyl. Inches.	Cut-off l. p. cyl. Per cent.	Valve travel h. p. Inches.	Valve travel l. p. Inches.	Outside lap h. p. Inches.	Outside lap l. p. Inches.	Inside lap h. p. Inches.	Inside lap l. p. Inches.	Inside clearance h. p. Inches.	Inside clearance l. p. Inches.	Length of port h. p. Inches.	Length of port l. p. Inches.	Maker's estimate of equal single expansion cyls. Ins.	Kind of service. Passenger or Freight.	Diameter of drivers.	Loco. Builder's standard cut-off for comps., h. p. cyl.	Loco. Builder's standard cut-off for comps., l. p. cyl.	Loco. Builder's standard ratio of receiver capacity to capacity of h. p. cyl.
Baldwin Decapod, Vauclain 4 cyl. for New York, Lake Erie & Western R. R.	16	27	28	25 3/8		26		6	6	7/8	5/8	0	0			28 1/2	28 1/2		Frgt.	50			
				25		25 5/8																	
				24		24 7/8																	
				23		24																	
				22		23 1/8																	
				21		22 3/8																	
				20		21 5/8																	
				19		20 3/4																	
				18		20																	
				17		19 1/8																	No receiver.
				16		18 3/8																	
				15		17 5/8																	
				14		16 3/4																	
				13		15 15/16																	
				12		15 1/16																	
				11		14 1/8																	
				10		13 1/8																	
				9		12 1/4																	
				8		11 1/8																	
				7		10																	
				6		8 7/8																	
				5		7 3/4																	

TABLE V.

Showing the Change in Exhaust Closure Affected by Using Inside Clearance or Negative Lap. Taken from a Schenectady 10 Wheeler on the Michigan Central R. R. 19 in. × 24 in. Single Expansion Engine and 20 and 29 × 24 in. Compound Engine. See Appendix R.

Negative Lap or Clearance, Inches.	Cut-off, per cent.	Compound.			Single Expansion.		
		Release of Steam per cent. of Stroke.	Compression of Steam per cent. of Stroke.	Valve Travel and Outside Lap, Inches.	Release of Steam, per cent. of Stroke.	Compression of Steam per cent of Stroke.	Valve Travel and Outside Lap, Inches.
⅛	33	74.4	80.0	6½	....	....	....
	41.7	78.1	82.9		....	....	....
	50	81.2	85.4		....	....	....
	58.5	84.4	88.5	1⅛	....	....	....
	83.5	94.8	96.3		....	....	....
¼	33	70.8	82.3	6½	62.5	80.2	5½
	41.7	73.5	85.4		66.7	82.3	
	50	78.1	87.5		73.9	87.5	
	58.5	81.2	89.6	1⅛	79.2	90.6	⅞
	83.5	93.7	96.9		92.7	96.9	
⅜	33	67.1	86.1	6½	....	....	....
	41.7	70.8	88.0		....	....	....
	50	75.5	90.1		....	....	....
	58.5	79.7	92.2	1⅛	....	....	....
	83.5	92.9	97.7		....	....	....
½	33	63.0	88.0	6½	....	....	....
	41.7	67.7	90.1		....	....	....
	50.	72.9	92.2		....	....	....
	58.5	77.6	93.8	1⅛	....	....	....
	83.5	91.9	97.9		....	....	....

CHAPTER XII.

MAIN VALVES.

77. **Lap, Travel and Size of Ports.**—The dimensions of the steam ports, valve travel, and outside and inside lap, suitable for compound locomotives, do not differ much from the best practice for single expansion locomotives, but it has been abundantly proved that better valve motions are needed for compound locomotives than are ordinarily used for single expansion. Also the valves and ports should be always in proportion to the cylinders, and this gives to the valves of the l. p. cylinders very large dimensions. The largest port in common use a few years since was 19 inches. Now the l. p. cylinders of compound locomotives have ports 24 inches long. Probably the compulsory use of longer ports and larger valves has had more to do with the recent tendency to use piston valves than any other factor. Large slide valves of the ordinary *D* form are very difficult to balance satisfactorily, and they cause a much increased wear on the eccentrics and links.

78. **Piston Valves.** — Piston valves are necessarily balanced from the nature of their construction, and certainly have been shown to be quite as applicable to locomotive work as to marine work, where they are now so commonly used. With a piston valve a very long port is readily obtained, and in fact a larger port is necessary, as the same length of port on the circumference of a piston valve is not as effective as a rectilinear port of the ordinary form with a flat valve.

Two express locomotives with piston valves have been built by Mr. von Borries for the German State Railroads. The experience with these engines shows that a piston valve must be considerably longer in circumference, which is in reality the length of the port, than is required with a flat valve to give equally good admission of steam. Ample room must be provided for the approach of steam to a piston valve or the advantage of its longer port will not be gained. Piston valves should have the same travel, and inside and outside lap as the ordinary form of *D* slide valves. The piston valve is, in fact, only a slide valve rolled up to form a cylinder, and needs the same treatment in design.

79. **Some Effects of Inadequate Valve Motions.**—The greatest evils which have to be met in arranging steam valves for compound locomotives are those of wire-drawing and compression. The wire-drawing in the h p. cylinder is practically no worse, nor more detrimental, than in a single expansion engine, but wire-drawing into the l. p. cylinder causes additional loss, and interferes with the adjustment of the power between the cylinders by means of the cut-off. In some compounds already built the wire-drawing through the main valve for the l. p. cylinder is so great that the cut-off point is not perceivable on the indicator card, and the engine works in about the same way as the old fashioned stationary engine with a throttle governor.

Compression causes more loss of power and efficiency in the h. p. than in the l. p. cylinder on account of the higher back pressure. In the l. p. cylinder the absolute back pressure at the time of exhaust closure is not far from 20 pounds, and with five compressions the terminal pressure at the end of the stroke would be not far from 100 pounds absolute. But in the h. p. cylinder the absolute back pressure is ordinarily about 65 pounds and with five compressions the pressure at the end of the stroke in the h. p. cylinder would be nearly 300 pounds, or very much above boiler pressure, 6. What actually does occur is this : when compound locomotives with the ordinary valve gear are running at a short cut-off and at high speed, the compression in the h. p.

cylinder rises to a point above the boiler pressure, where it lifts the main valve, and the excess of steam in the clearance spaces and ahead of the piston is pushed into the steam chest. This will be observed in Figs. 11, 12, 14, 15, 112, 113, 127, 136 and 149.

Whether a piston valve, or a slide valve of the ordinary kind is used, the simplest way to reduce wire-drawing and compression after making the ports as long as is practicable, is to increase the valve travel, increase the outside lap, and cut out the valve on the inside to give what is called "clearance" or "negative" lap. See Table U1.

The effect of increasing the valve travel and outside lap, is to give a greater port opening at short cut-offs and to postpone the point of compression toward the end of the stroke, thus reducing compression. The effect of cutting out the inside of the valve to make a negative lap is to delay the closure of the exhaust and reduce compression.

80. **Effect of Long Valve Travel and Inside Clearance or Negative Lap.**—The following will illustrate the benefit obtained from a change in valve travel, outside lap, and from the use of inside negative lap: A 5½ inch travel with ¾ outside lap will give about $\frac{5}{16}$ inch port opening at 25 per cent. cut-off. A 7 inch travel and 1¼ inches outside lap will give nearly ½ inch port opening at the same cut-off. ⅜ inch negative lap will reduce compression to a point somewhat below the admission pressure, which is where it should be, when used on an engine which formerly had positive inside lap, and a compression much above boiler pressure before the completion of the stroke. 7 inches valve travel on a locomotive is not so great as to lead to any mechanical difficulties in operation or design. This has been conclusively shown by the experience of Mr. L. B. Paxson, S. M. P., of the Philadelphia & Reading Railroad, Figs. 39 to 42, and by the experience of the Rhode Island Locomotive Works. These two

companies have led in this country in the matter of long valve travel. As much as ½ inch negative lap on each side on the h. p. cylinder has been used with success on high speed compound locomotives. ⅜ inch negative lap on each side has been used with success on the l. p. cylinder. $\frac{5}{16}$ inside negative lap has been used with excellent results on single expansion locomotives, and the experience already had shows beyond doubt that inside clearance is absolutely necessary on all high speed locomotives, whether single expansion or compound, if the best results are desired. It is practically impossible to design a high speed compound locomotive, no matter what the type, that will run without excessive wire-drawing and compression with the Stephenson link motion or with any of the commonly used locomotive valve gears, without using a long valve travel and considerable inside clearance or negative lap for both cylinders. The greater amount of negative lap is needed for the h. p.cylinder.

The effect of inside clearance or negative lap on steam distribution at various low speeds, and the effect it has on the shape of indicator cards, is shown by Fig. 43, indicator cards Nos. 1 to 6. The data for these cards is given in Table W. It will be noticed that at low speeds the steam from the exhaust of one end of the cylinder passes over into the other end of the cylinder through the opening that is made between the two ends of the cylinders by the use of negative lap. The negative lap in this case was $\frac{5}{16}$ and $\frac{3}{16}$ inches. From card No. 6 it is clear that this transfer of steam, at the time of exhaust from one cylinder to the other, disappears almost entirely when speed has increased to 32.5 miles per hour. These cards also show that the engine from which they were taken had liberal steam pipes and passages, as the steam chest pressure and receiver pressure varies but little from the pressure at admission. These are admirable cards from a compound locomotive for the speed at which they were taken.

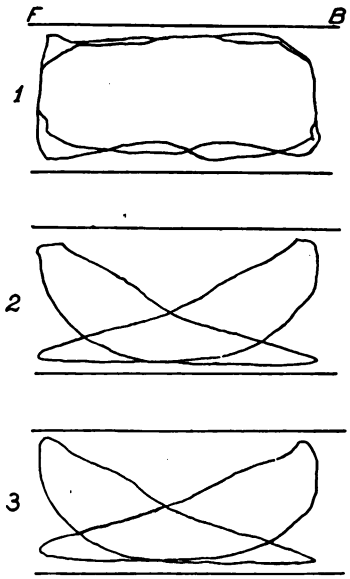


FIG. 39.
$\frac{1}{64}$ Inch Inside Lap.

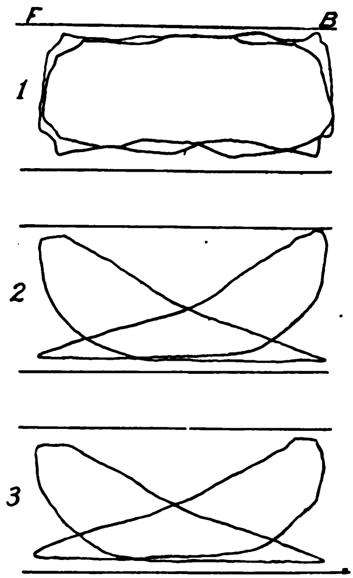


FIG. 40.
$\frac{1}{4}$ Inch Negative Lap.

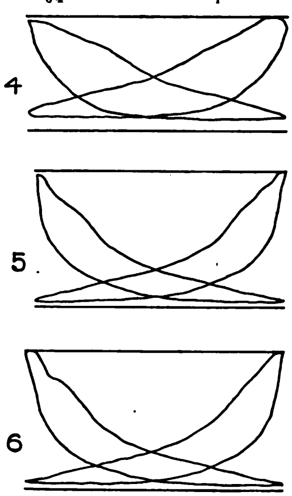


FIG. 41.
$\frac{1}{64}$ Inch Inside Lap.

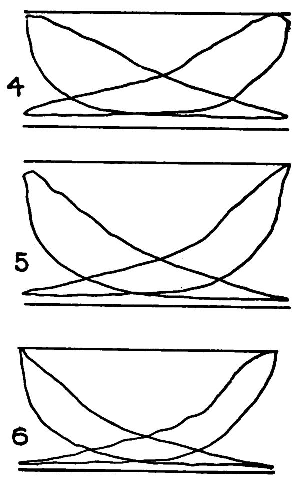


FIG. 42.
$\frac{1}{4}$ Inch Negative Lap.

Indicator Cards Showing the Effect of Negative Lap.

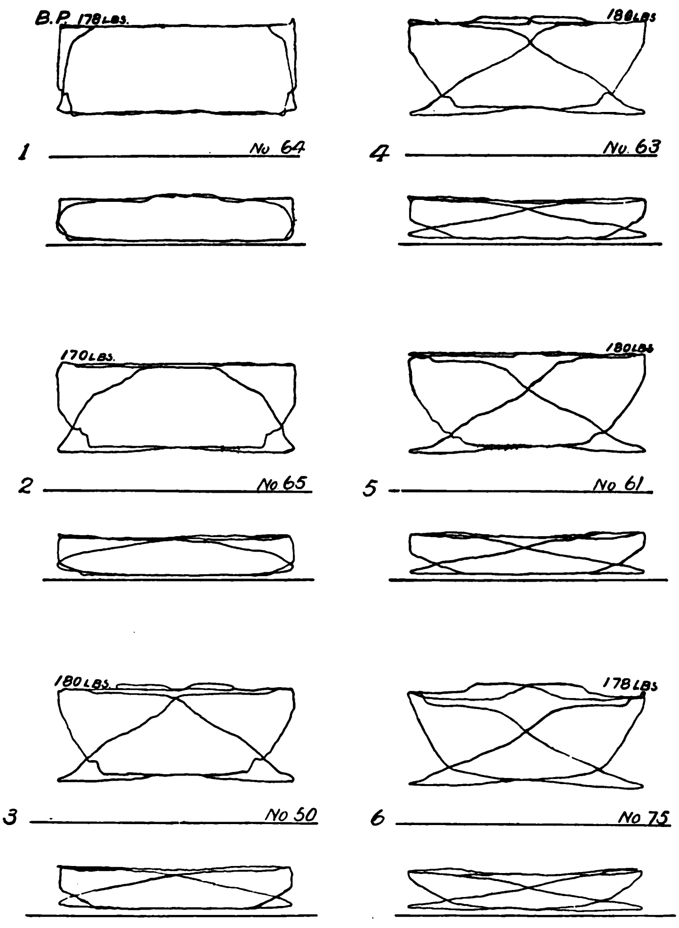


FIG. 43.

Indicator Cards from Compound Locomotive Showing Effect of Negative Lap at Low Speed.

TABLE W.

Giving Data about Indicator Cards Nos. 1 to 6, Fig. 43, from Schenectady (Pitkin) Compound on the Michigan Central Railroad. See Appendix R.

Number of Card.	Revolution per minute.	Miles an hour.	Cut-off in Inches. H. P.	Cut-off in Inches. L. P.
1	40	6.8	21⅞	22¼
2	72	12.2	17	18⅛
3	104	17.6	13½	15⅛
4	108	18.3	12	13⅛
5	104	17.6	10¾	12⅜
6	192	32.5	10¾	12⅜

Length of Valve Travel - - - - 6½ in. full gear.
" " Steam Ports, l. p. cyl. - - - - - - 20"
" " " " h. p. cyl. - - - - - - 18"
Width of " " l. p. cyl. - - - - - - 2⅛"
" " " " h. p. cyl. - - - - - - 2⅛"
Outside Lap, h. p. cyl. - - - - - - 1⅛"
Inside Clearance, h. p. cyl. - - - - - - $\frac{5}{16}$"
Outside Lap, l. p. cyl. - - - - - - 1⅛"
Inside Clearance, l. p. cyl. - - - - - - $\frac{3}{16}$"

TABLE X.

Giving data regarding Figs. 39 and 41, cards Nos. 1 to 6, taken from a Philadelphia & Reading express engine, with Single Expansion cylinders.

Card Number.	Revolutions per Minute.	Miles per Hour.	Cut off. Inches.	Mean Effective Pressure. Front.	Mean Effective Pressure. Back End.	Inside Lap of Valve. Inches.	Inside Clearance of Valve. Inches.	Position of Throttle.
1	136	27.7	Full stroke.	106.8	109.35	$\frac{1}{64}$	....	Wide open.
2	262	53$\frac{38}{100}$	6½	45.75	50.4	$\frac{1}{64}$	....	" "
3	280	57	6½	49.2	55.8	$\frac{1}{64}$	....	" "
4	294	59$\frac{88}{100}$	6½	39.45	44.1	$\frac{1}{64}$	....	" "
5	282	57$\frac{44}{100}$	4	33.19	......	$\frac{1}{64}$	....	" "
6	294	59$\frac{88}{100}$	4	32.65	35.55	$\frac{1}{64}$	....	" "

The small effect of a little inside clearance or negative lap is very clearly and satisfactorily shown by Figs. 39, 40, 41, and 42, indicator cards Nos. 1 to 6, which were taken from a 21 × 22 inch Philadelphia & Reading express loco-

TABLE Y.

Giving data regarding Figs. 40 and 42, cards Nos. 1 to 7, taken from a Philadelphia & Reading express engine, with single expansion cylinders, and showing the small effect produced by a little inside clearance or negative lap, also showing the need of much inside clearance, and showing also the slight effect on steam distribution produced by inside clearance at slow speed.

Card Number.	Revolutions per Minute.	Miles per Hour.	Cut-off. Inches.	Mean Effective Pressure. Front.	Mean Effective Pressure. Back End.	Inside Lap of Valve. Inches.	Inside Clearance of Valve. Inches.	Position of Throttle.
1	119	24	Full stroke.	110.10	107.25	....	¼	Wide open.
2	264	$53\frac{77}{100}$	6½	55.8	48.	....	¼	" "
3	282	$57\frac{44}{100}$	6½	53.25	54.75	....	¼	" "
4	294	$59\frac{88}{100}$	6½	43.5	42.75	....	¼	" "
5	318	$64\frac{87}{100}$	4	42.6	38.55	....	¼	" "
6	324	$65\frac{99}{100}$	4	36.15	26.40	....	¼	" "
7	348	$70\frac{88}{100}$	6½	45.	44.4	....	¼	" "

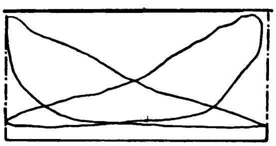


FIG. 44.

Good Steam Distribution in Single Expansion Locomotive Cylinders.

motive. The normal boiler pressure is 145 pounds by gauge. Tables X and Y give the data about these cards. This engine is unusual in having 7 inches valve travel and 1¼ in. outside lap. Probably these indicator cards are the best ever taken from a locomotive with a Stephenson link motion. The good points of such a card as No. 7, Table Y, Fig. 44, taken from this engine with ¼ inch negative lap, at 70 miles an hour show how difficult it will be for a compound locomotive to make a saving in passenger service against a locomotive having long valve travel and inside clearance and ample area of steam and exhaust ports. The effect of using inside clearance on the distribution of steam at slow speed is very small, as is clearly shown by a comparison of Figs. 39 and 41, with Figs. 40 and 42. These cards, taken with the throttle full open, show that considerable inside clearance is needed to produce a substantial benefit at high speed with long valve travel. The ¼ inch inside clearance used on this engine would have shown a more substantial change in mean effective pressure at high speed if the valve travel had been 5½ inches, as is common on locomotives. The new compound locomotive designed by Mr. Axel S. Vogt, Mechanical Engineer of the Pennsylvania Railroad, has ¾ inches inside clearance in the h. p. cylinder and ⅝ inches in the l. p. valve, and 7 inches valve travel.

81. **Conclusions about Main Valve Dimensions.**—It is impossible to give a general rule for the area of steam ports and valves of compound locomotives. What can be said that is useful is: the valve travel should be as long as it can be made without inducing mechanical difficulties. It probably should never be less than 6 inches and would better be 7 inches. The inside clearance or negative lap should be what is necessary to reduce compression, and the increase of the negative lap should be carried on until the indicator cards from the engines under normal conditions are comparatively satisfactory in the matter of compression. If there is any waste due to negative lap, it will show on the indicator card, and can be estimated therefrom and in no other way except by an elaborate shop test. There is a tradition among railroad men which has operated against the proper use of negative lap. This tradition is to the effect that inside clearance causes a waste of steam, but this tradition is not founded on fact and is true only for very slow speed locomotives.

In conclusion, about all that can be said to assist the designer is that the best modification that can be made of the common form of link motion will be none too good for compounds, and it is well worth while to pay a considerable sum to get long valve travel and large steam ports. If the steam ports are made the same length for both h. p. and l. p. cylinders and are made considerably longer than the diameter of the h. p. cylinders, say 20 per cent. and the valve travel for the common sizes of locomotives is made from 6½ to 7½ inches, and the negative lap about ¼ inches for freight and ⅜ inches for passenger for the l. p. cylinder, and ⅜ of an inch for freight and ½ inches for passenger for the h. p. cylinder, the practical results from service will be pretty nearly satisfactory. If the compression with these proportions is too great, as it may be for high speeds, the negative lap must be increased. If there is too much wire-drawing, there is only one recourse with the Stephenson link, namely, to increase valve travel and the length of the ports. Of course, the supplementary or Allan port is an advantage, but does not give as much benefit as an increase of valve travel. To add an Allan port to long travel valves sometimes causes trouble in design, as the valve must be longer and the area to be balanced will be larger. Allan ports for piston valves are scarcely practical, as the same effect can be gained by making the valve with a larger diameter, and this is easier and simpler than to introduce the additional packing rings for the Allan port. In these remarks all consideration of the unusual forms of valve gears so far tried have been omitted for the reason that none of them have proved to be what is wanted for practical work.

CHAPTER XIII.

STEAM PASSAGES—ACTION OF EXHAUST.

82. **Size of Steam Passages and Loss Due to Wire-Drawing.**—All of the rules applicable to the use of steam, so far as steam passages are concerned, that are common in stationary engine work apply with equal force to locomotives. Formerly it was a common defect in locomotives to have too small steam passages, but now a few of the modern designs have the same area of passages as are provided for stationary engines. When this is done, and the engine is run with a wide-open throttle, the difference in pressure between the boiler and the steam chest will be very small, even when the locomotive is running at considerable speed. See Figs. 43, 46 and 47.

The loss due to running a locomotive with a partly open throttle or with too small steam passages is more than is generally understood, as has been recently proved by the shop tests of a locomotive by Professor Goss at the Purdue University. Diagram, Fig. 45, shows in a general way what these results indicated. Such wire-drawing as is shown on the diagram gave considerable super-heat to the incoming steam, but the reduction in cylinder condensation due to super-heat did not offset the loss in potential of steam pressure. Therefore, it may be concluded that locomotives should have large steam passages and be run with a wide-open throttle; more particularly is this true of the compound locomotive, which depends for its economy upon the utilization of the high potential of increased steam pressure by giving greater expansion.

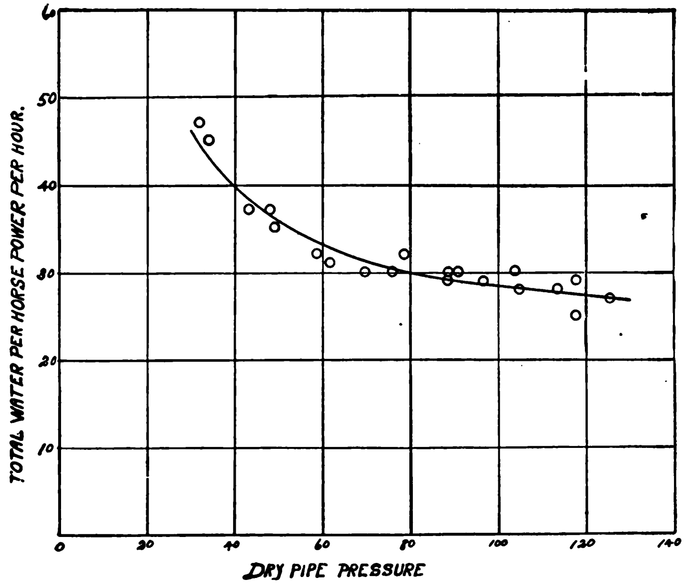


FIG. 45.
Diagram Showing Loss of Efficiency Due to Wire-Drawing Through Throttle.

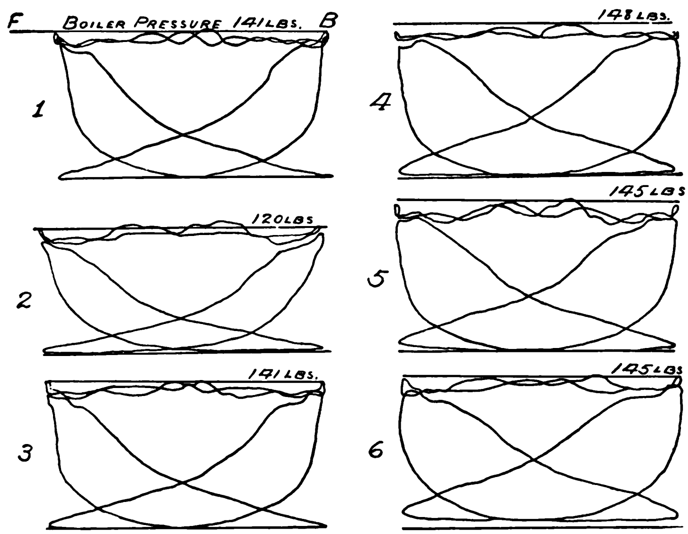


FIG. 46.
Indicator Cards Showing Difference Between Boiler, Steam Chest and Initial Pressures.

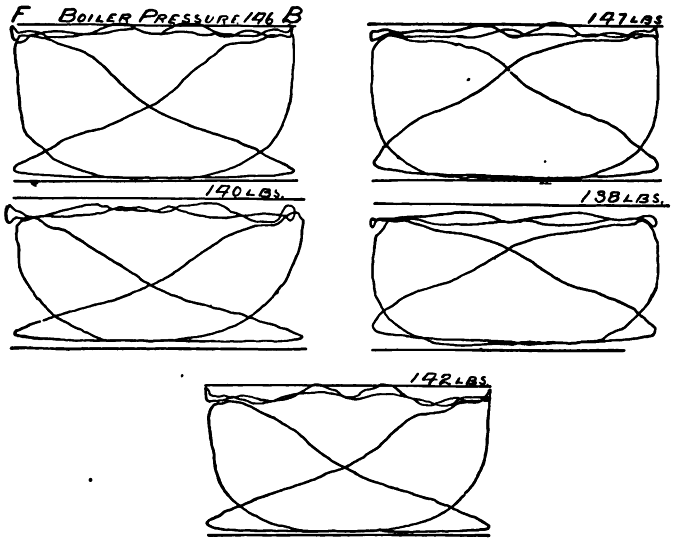


Fig. 47.

Indicator Cards Showing Difference Between Boiler, Steam Chest and Initial Pressures.

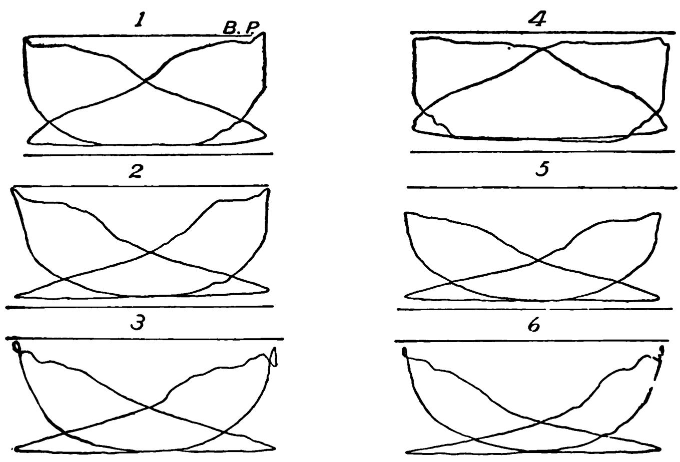


Fig. 48.

Indicator Cards Showing Effect of Small Nozzles on Back Pressure. See Table AA.

The variation in steam chest pressure of locomotives, where the throttles are of proper dimensions, and the steam pipes and passages are adequate, is shown by Figs. 46 and 47, Cards Nos. 1 to 11, Table Z, which were taken from a 16 × 24 passenger engine on the Chicago, Milwaukee and St. Paul road. The small drop between the boiler and the

TABLE Z.

Showing the Variation in Steam Chest Pressure on a Single Expansion Locomotive, illustrating the comparatively small drop between the Boiler and the Steam Chest at a speed not exceeding 45 miles per hour, when the throttle is wide open. See Figs. 46 and 47.

Card Number.	Cut-off, Front End. Inches.	Cut-off Back End. Inches.	Mean Effective Pressure, Front End. Pounds per sq. in.	Mean Effective Pressure, Back End. Pounds per sq. in.	Revolutions per Minute.	Miles an Hour.	Total Indicated Horse Power.	Boiler Pressure by Gauge.
1	$2\frac{7}{8}$	3	37.4	42.0	132	24.2	254	141
2	$6\frac{5}{16}$	$4\frac{3}{8}$	29.4	29.2	240	44.2	338	120
3	$5\frac{7}{16}$	$5\frac{1}{2}$	42.6	45.4	172	31.7	362	141
4	$6\frac{3}{4}$	$7\frac{1}{4}$	55	58.4	152	28	412	148
5	$6\frac{3}{4}$	$7\frac{1}{4}$	51	54.2	168	31	422	145
6	8	$8\frac{1}{2}$	55.2	57.4	192	35.4	518	145
7	8	$8\frac{1}{2}$	65.6	68.	148	27.3	479	146
8	8	$8\frac{1}{2}$	49	51.	204	36.5	489	140
9	$9\frac{3}{8}$	10	57.6	61.	140	25.6	398	142
10	$10\frac{13}{16}$	$11\frac{7}{16}$	85	88.4	108	20	448	147
11	$12\frac{1}{8}$	$12\frac{9}{16}$	73.8	74.8	168	31	599	138

steam-chest in this case is due to a wide-open throttle. At high speeds the drop increases somewhat depending upon the cut-off and the size of the passages, but these cards show what may be expected in fairly well designed locomotives at a speed not exceeding 240 revolutions per minute, which in this engine amounts to about 44 miles per hour. The engine is a 16 × 24 inch cylinder, five-foot wheel passenger locomotive of the eight-wheel American type.

83. **Effect of Exhaust on Fire and on Back Pressure.**—The lower pressure and greater volume of the exhaust from the compound locomotive appears to produce

a more uniform and a better effect on the fire. This advantage, added to the decrease in the total fuel consumption per minute, resulting from the saving of the compound, has, so far as can be seen, caused a secondary saving of fuel due to compounding. There are, then, perhaps, two savings due to compounding. The primary saving due to the compounding *per se*, and the secondary resulting from the better action of the draft and the decreased forcing of the fires, 142.

In suburban or elevated railroad service, where mufflers are put on the exhaust pipe of single expansion engines to decrease the noise of the exhaust, the use of the compound locomotive, with its lower pressure of exhaust, enables the mufflers to be dispensed with. In this way as much as 20 per cent. of steam may be saved by the reduction of the back pressure in the cylinders caused by the mufflers. Mufflers clog up quickly and have to be bored out frequently, or the back pressure becomes so great as to make the engines "logy," 139–147.

TABLE AA.

Back Pressure Before Changing Valves and Nozzles.

No. of Card.	Speed. Miles per hour.	Cut-off.	Boiler Pressure. Pounds.	Initial Pressure. Pounds.	Mean Effective Pressure. Pounds.	Mean Back Pressure including Compression. Pounds.
4	15.	12″	160.	151.	98.	20.5
1	24.	9″	160.	150.	79.	22.5
2	30.	8″	160.	143.	57.	25.5
5	35.	8″	158.	135.	51.	27.
3	42.	8″	160.	150.	57.	31.5
6	53.	5″	160.	146.	39.	28.

Figs. 48 and 49 show the need of very carefully watching the details of a new design of engine, by examining indicator cards, to prevent losses in the cylinders by back pressure. Fig. 48, Indicator Cards Nos. 1 to 6, see Table AA, gives the back pressure in a ten-wheel engine with a 3⅛ exhaust nozzle double, that is, with a separate nozzle for each cylinder. The nozzles were increased in diameter $\frac{1}{8}$ of an inch, and the inside lap was cut out from $\frac{1}{16}$ on both sides to $\frac{1}{32}$ negative lap on both sides. The decided reduction in back pressure, as shown by Fig. 49, Cards Nos. 1 to 6, and by Table BB, changed the engine

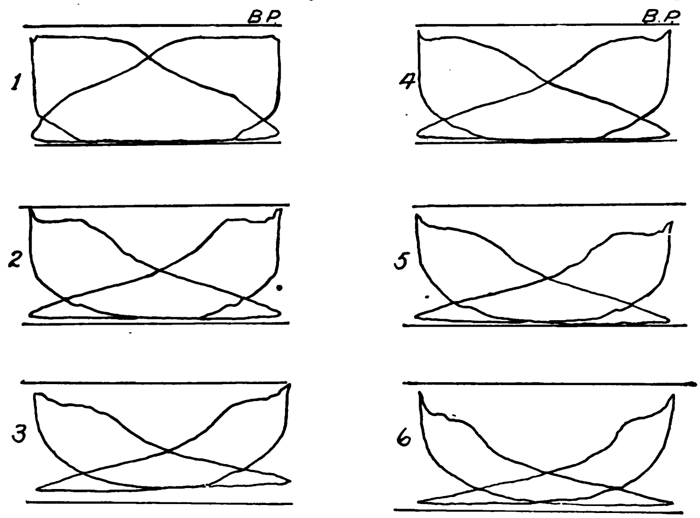


FIG. 49.
Indicator Cards Showing Decrease of Back Pressure Following an Increase of the Diameter of Exhaust Nozzle. See Table BB.

TABLE BB.

Back Pressure after Changing Valves and Nozzles.

No. of Card.	Speed, Miles per hour.	Cut-off.	Boiler Pressure, Pounds.	Initial Pressure, Pounds.	Mean Effective Pressure. Pounds.	Mean Back Pressure including Compressions. Pounds.
1	18	10″	158	145.	98.	9.5
4	24	9″	160	148.	75.	16.
2	29	8″	158	143.	66.	17.5
5	33	8″	155	138.	58.	18.5
3	42	8″	155	137.	52.	24.5
6	54	5″	155	130.	40.	23.

materially. The difference was enough to make quite a saving in fuel. Such a change as this in back pressure produces the effect on an engine that is known to locomotive engineers as "smarter," that is, the engine has a livelier action.

Fig. 49a illustrates how the mean effective pressure is effected by an increase or decrease of back pressure. The two sets of cards shown are those numbered 5 in Tables AA and BB. The space between the cards that is sectioned by vertical lines shows the change in mean effective pressure brought about by a variation in the back pressure. It is evident from this illustration that the effect of an increase or decrease of back pressure extends over the entire

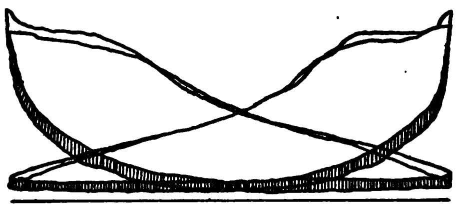


FIG. 49a.

Effect of Back Pressure on Mean Effective Pressure.

length of the indicator card. The reason of this is that when the back pressure is increased or decreased the pressure at the commencement of compression is correspondingly increased or decreased, and the whole compression curve is therefore effected. The conclusion from an examination of Figs. 48, 49 and 49a must be that a careful selection of exhaust and draught apparatus is necessary in order to produce an economical and powerful engine at high speed.

CHAPTER XIV.

EFFECT OF HEAVY RECIPROCATING PARTS.

84. **Weight of Reciprocating Parts.**—It is of the utmost importance that the weight of the pistons, crossheads, piston rods, main rods, and in fact all the reciprocating parts of a locomotive be kept down to the lowest limit. The reason is that these parts have to be balanced to make the engine ride steadily, and this balance acts at all points of a revolution of the drivers. It has an outward or centrifugal tendency from the centre of the wheel that is very great at high speed. This tendency of that part of the balance that is used for the reciprocating parts is counteracted only when the balance is in the horizontal position, that is, when the crank is at the end of the stroke. At other times the centrifugal tendency is upward or downward, and is unresisted except by the rail or the springs above the axle boxes. This centrifugal tendency is sometimes so great as to lift the wheel from the rail. And it has in some cases seriously damaged the tracks during a single run by a badly balanced locomotive at high speed. It is then necessary to reduce the weight of the reciprocating parts, and thereby the reciprocating balance as much as possible. Unfortunately compound locomotives carry with them a necessity for larger pistons. These large pistons will of course be heavier than smaller ones, but are not necessarily heavier than those that are now commonly used here for single expansion engines. In the United States builders are much behind European practice in piston and crosshead construction. The weight of the reciprocating parts used here is more than twice as great

as those used in Europe for the same size of cylinder. This results from the use here of a cheaper type of piston. The foreign type is generally of forged steel with a single plate. Here they are generally made of cast iron with double plates. By using some of the higher grades of manganese steel, or aluminum bronze, or by using forged steel, the reciprocating parts of either a two-cylinder receiver compound or a four cylinder non-receiver compound would not weigh more than the reciprocating parts of some of our present single expansion engines.

A commendable step that has been taken in the reduction of reciprocating parts is the removal of the non-useful weight in the Vauclain crosshead by the Baldwin Locomotive Works. This is shown in Fig. 119. This crosshead is made of cast steel and cored out to remove all weight possible. Such reduction of weight is possible in all American types of crossheads, and a similar reduction is possible with American pistons. By devoting as much attention to reduction of weights and reciprocating parts as the matter deserves, the total weight might be reduced at least 50 per cent.

85. **Advantage of Large Drivers.**—Large drivers reduce the number of revolutions per minute, and thereby decrease not only the piston speed, but also the effect of the counterbalance weight, and therefore a large wheel is advantageous for a compound, as it reduces the wire-drawing and compression in the cylinders, and decreases the effect of reciprocating parts. See Fig. 50.

86. **Counterbalancing of Reciprocating Parts.**—Counterbalancing is a matter that requires especial attention in selecting a compound. Only the lightest practical reciprocating parts should be used, and the practice followed in high speed marine work will serve as a guide.

87. **Marine Practice in Counterbalancing.**—There are large triple expansion marine engines running at piston

speeds of over 800 feet per minute. The piston speed attained by the quadruple expansion engines of the torpedo

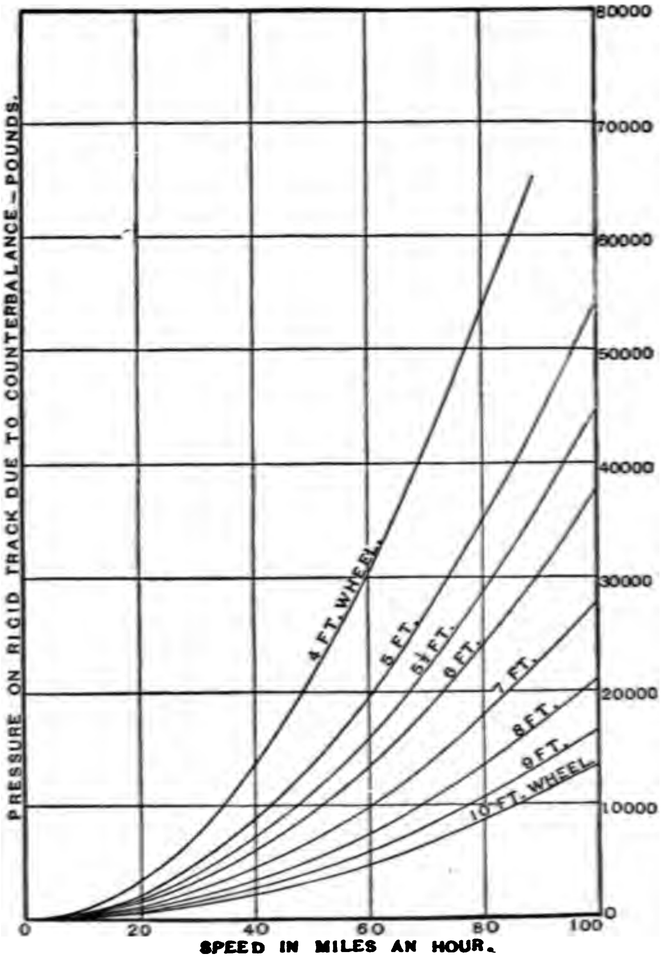


FIG. 50.

Diagram Showing Decrease in Pressure on Track Due to Counterbalance Which Follows an Increase in the Diameter of Drivers.

boat "Cushing" was 925 feet per minute on her trial, and the speed of pistons of the triple expansion engines of a recent Turkish torpedo boat is given as 936 feet per minute on a trial trip. If these speeds are practicable with triple and quadruple expansion engines, there does not appear to be any good reason for doubting the practicability of speeds of 1,100, or even 1,400 feet, with compound locomotives. There is undoubtedly a maximum limit to piston speed, and it is lower for compound engines than for single expansion engines, but the limit is sufficiently high to be comparatively unimportant to the designer of locomotives. The principal factor which limits the speed is the weight of the reciprocating parts. In an engine working at a speed of 250 revolutions per minute, the reciprocating parts must be started from a state of rest at the beginning of each stroke, and their speed accelerated to about 26 feet per second during approximately a half stroke, which occupies about 0.06 second. A very full and complete discussion of this subject will be found in a paper by Mr. D. S. Jacobus, in Vol. XI. of the Transactions of the American Society of Mechanical Engineers. See Appendix P.

The pressure per square inch of piston, for a locomotive having a cylinder 18½ inches in diameter and 24 inches stroke, required to overcome the inertia of the reciprocating parts and accelerate them at 250 revolutions per minute, varies from about 55 pounds at 10 degrees from the dead point to 0 at about 80 degrees. The work stored in the reciprocating parts during the first half of the stroke is, of course, transmitted to the crank pin during the last half of the stroke. But the effective pressure on the crank pin during the first half stroke is only that due to the difference between the apparent pressure as shown by the indicator card and that necessary to accelerate the reciprocating parts. It is evident that if the pressure of the steam on the piston is just equal to that required for acceleration at any position of the piston, no pressure will be transmitted to the crank pin at that point in the stroke, and that if these pressures are equal during the period of acceleration, all pressure which is transmitted to the crank pin during the stroke will be during the second half stroke.

The pressure necessary to produce acceleration varies directly as the weight of the reciprocating parts, and as the square of the speed of rotation. The possible means of reducing this pressure are therefore to make the reciprocating parts lighter, or the driving wheels of greater diameter

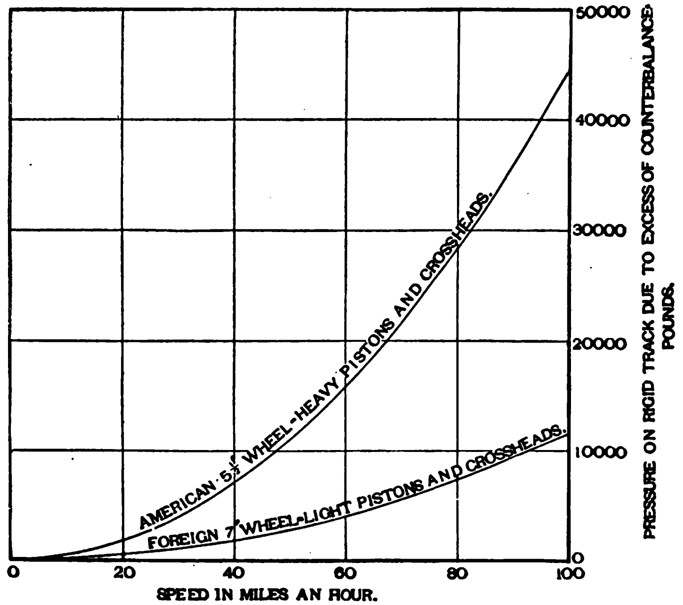


FIG. 51.

Diagram Showing Difference in Counterbalance Pressure on Track in American and Foreign Engines.

so as to reduce the speed of rotation. How much the distribution of pressures on the crank pins will be affected by such changes is a question which must be solved by the designer in each case, and it is a factor which is worth careful consideration, more on account of the crank-pin pressures than on account of the limitations of speed. A considerable reduction in weight is effected by the use of steel, wherever practicable, for the reciprocating parts, and the adoption of the most economical shapes for connecting and coupling rods, pistons and cross-heads.

88. **Effect of Decreasing Weight of Reciprocating Parts and Increasing Diameter of Drivers.**—Fig. 51 gives the maximum pressure on a rigid track due to that portion of the counterbalance of a locomotive that is used to counteract the horizontal effect of the reciprocating parts of American and foreign locomotives of the same size of cylinder. This diagram also illustrates the reduction of the variations of pressure of driving wheels upon the track

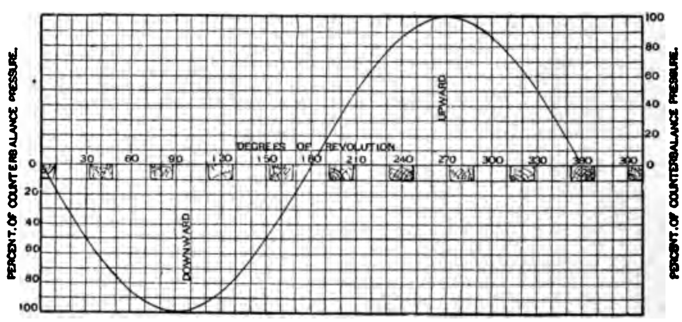


FIG. 52.

Diagram Showing Distribution of Counterbalance Pressure over Track and the Per Cent. of Maximum Centrifugal Pressure Which Occurs at Different Points of a Revolution.

that follows an increase in the diameter of the driving wheels and a reduction of the weight of the reciprocating parts.

Fig. 50 shows the advantage of a large wheel in reducing the centrifugal tendency. The pressures given are all calculated for an engine with an 18 × 24 cylinder

89. **Distribution of Centrifugal Tendency of Counterbalance over the Track.**—Fig. 52 shows the variation in the per cent. of the maximum centrifugal tendency of counterbalance, which is exerted on the track at different

points during a complete revolution of the driving wheels when the track is rigid and does not deflect under the load due to the centrifugal tendency. It also shows how the maximum track pressure is distributed over several ties, and how it gradually increases and decreases.

CHAPTER XV.

DESCRIPTION OF TWO-CYLINDER RECEIVER COMPOUNDS WITH AUTOMATIC INTERCEPTING VALVE STARTING GEARS, AND WITHOUT SEPARATE EXHAUST FOR HIGH-PRESSURE CYLINDER AT STARTING.

The inauguration of the present era of compound locomotives in Europe is due to Mr. Anatole Mallet, who designed successful two-cylinder compound locomotives for the Bayonne & Biarritz Railroad in 1876, and has since brought out many different designs. While it would not be incorrect to class the greater number of compound locomotives as belonging to the Mallet system, this term as applied to two-cylinder engines is usually restricted to those which can be operated either as single expansion or compound engines at the will of the engineer (non-automatic) as distinguished from those which are necessarily worked as compound engines, except for a brief interval in starting (automatic).

The disposition of cylinders and steam chests with regard to the boiler and running gear of two-cylinder compound locomotives does not differ from the practice in single expansion locomotives. The same diversity of design that has heretofore been remarkable in European practice as compared with the American, is found in compound locomotives. The designer will find precedent in existing engines for almost any arrangement of principal parts and for any type of valve gear which he is likely to adopt.

There have been quite a large number of inventions of somewhat minor value in the details of starting gear, more particularly of the automatic type, for two-cylinder compound locomotives, and a number of patents have been taken out in this and foreign countries, but as a rule they differ so little from the original designs of Mallet and von Borries that the patents are weak and the scope limited to some specific construction. It is impossible to give within the limits of these pages anything like a complete résumé of the art at this time as exhibited in the Patent Office. It is not useful to do so, as the reader would be confronted with a mass of drawings and descriptions which would lead to no conclusions. Only the principal designs and such as have actually been put into service are here described.

90. **The von Borries System in 1889.**—This system is strictly automatic, which means, that the change from the use of boiler steam in the l. p. cylinder to full compound action is made automatically without the will of the engineer, and takes place whenever the accumulated pressure of the exhaust from the h. p. cylinder in the receiver is sufficient to operate the automatic mechanism. Figs. 53 and 54 illustrate one of the arrangements of cylinders and steam connections in two designs of compound locomotives according to the von Borries system. In both figures *h* is the h. p. cylinder, *l* is the l. p. cylinder, *A* is the steam pipe from the boiler to the h. p. cylinder, *C* is the receiver connecting the two cylinders, *V* is the starting and intercepting valve, *B* is the auxiliary steam pipe from the boiler to the starting valve, and *D* is the exhaust pipe from the l. p. cylinder.

The essential feature of the von Borries system is the combined intercepting and starting valve, an early form of which is illustrated by Fig. 55. In this figure *a* is the receiver pipe which leads from the h. p. cylinder and *b* is the passage to the l. p. cylinder. The valve is shown in the position which it occupies ordinarily, or when the locomotive is working as a compound engine, the direction of the flow of the steam being as indicated by the arrows. Connected to the back of the intercepting valve *v* are two small plungers *c c* which together form the starting valve. Sup-

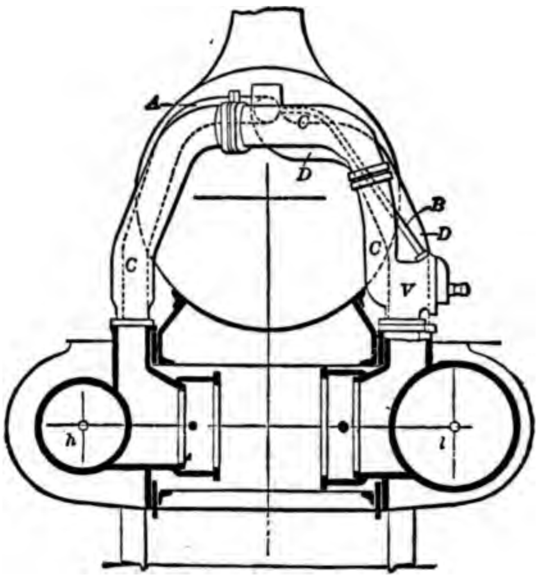


FIG. 53

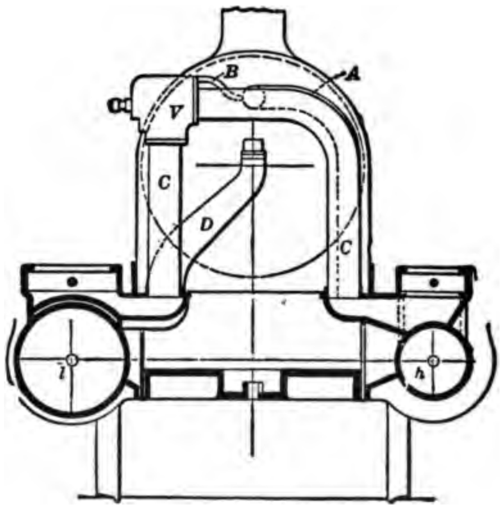


FIG. 54.
Arrangement of Cylinders and Intercepting Valve with von Borries Automatic Starting Gear.

posing the valves to be in the positions shown in Fig. 55 and the engine about to start, when the throttle is opened steam will be admitted to the h. p. cylinder by the usual pipe, and also to the auxiliary steam pipe *d*, and by the passage shown to the back of the plungers. The pressure on the ends of the plungers is sufficient to move the intercepting valve *v* to the left in the figure until it seats at *e*. By the same movement two small ports *h h* are uncovered, through which steam from the boiler is admitted to the passage *b* and thence direct to the l. p. steam chest, while, as the intercepting valve is closed, this pressure does not act against the h. p. piston.

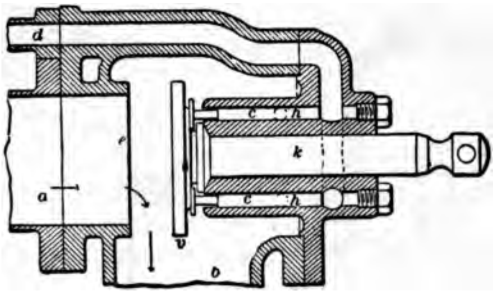


FIG. 55.
von Borries Intercepting Valve, Early Form.

As the engine starts and the exhaust from the h. p. cylinder takes place, the pressure in the receiver rises until it is sufficient to overcome the pressure on the l. p. side of the intercepting valve, when this valve is moved back to the position shown in the figure, while at the same time the two small steam ports are closed by the plungers, and the engine begins to work as a compound. It is said that in practice the pressure of the steam from the boiler which is admitted to the l. p. cylinder is reduced by wire-drawing, due to the small steam pipe and ports, to about one-half the boiler pressure, and as the ratio of the cylinders is about

2, the total pressure on the two pistons in starting is nearly equal. To prevent excessive pressure in the l. p. cylinder and receiver a safety valve is placed on the latter.

The pressure in the receiver when running is sufficient to overcome the boiler pressure acting on the ends of the two small plungers, together with the atmospheric pressure on the stem of the large valve *v*, and therefore the valves are maintained in the position shown in Fig. 55 as long as the engine is running under steam.

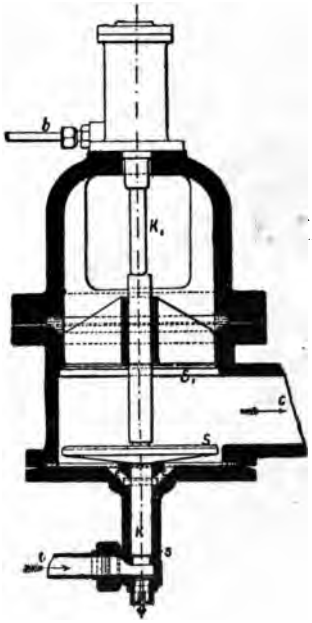


FIG. 56.

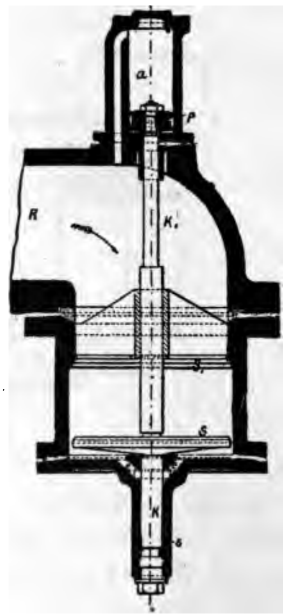


FIG. 57.

von Borries Intercepting Valve as Used on Jura, Berne-Lucerne Ry.

91. The von Borries System, as used on the Jura, Berne-Lucerne Railway.—To facilitate starting, the engine is fitted with a von Borries automatic starting valve, the construction of which is shown by the detail views, Figs. 56 and 57, annexed. This apparatus is placed at the junction of the intermediate receiver or connecting pipe with the l. p., the steam leaving this intermediate receiver at *R* and passing off to the l. p. cylinder at *C*. If the engine stops with the h. p. piston on a dead point, so that the engine cannot start in the ordinary way and no steam can be exhausted to the l. p. cylinder, the live steam passes from the h. p. valve chest through the pipe *t* and acts upon the lower end of the spindle *K*, the pressure thus exerted raising the valve *S* and closing it on the seat *S1*. When the spindle *K* is thus lifted it uncovers the small openings *e e*, and live steam can then pass to the l. p. cylinder, thus starting the engine. As soon as the engine gets to work the exhaust from the h. p. cylinder, of course, raises the pressure in the intermediate receiver, and this pressure acting on the valve *S* overpowers the pressure of the live steam on the lower end of the spindle *K* and the receiver pressure on the valve *S*, and forces the valve off its seat, thus allowing the exhaust steam from the h. p. cylinder to pass to the l. p., the engine then continuing to work compound. To insure the valve *S* being forced down into the position in which it is shown in Figs. 56 and 57, there is provided a small piston *p* working in a cylinder *a*, the upper end of which is in free communication with the receiver. The area of this piston is such that the pressure of the receiver steam on it is sufficient to over-power the pressure of the live steam on the lower end of the spindle *K*.

92. **A Modification of the von Borries System.**—A modification of the von Borries intercepting valve is shown in Fig. 58. This valve is placed in the side of the smoke box, and is connected at *A* by a small pipe to the steam pipe from the boiler. When the throttle is opened steam enters the passage *C* by way of the pipe, and pressing against the shoulder of the steel spindle *D*, pushes it into the position shown in Fig. 58, and thus closes the valve. The steam then passes around the spindle, out

through the ½ inch opening and into the chamber *B*, which communicates with the receiver. Then it has free access to the steam chest of the l. p. cylinder. It also acts against the piston *E* through the passage *F*, but the greater area of the main valve keeps it closed.

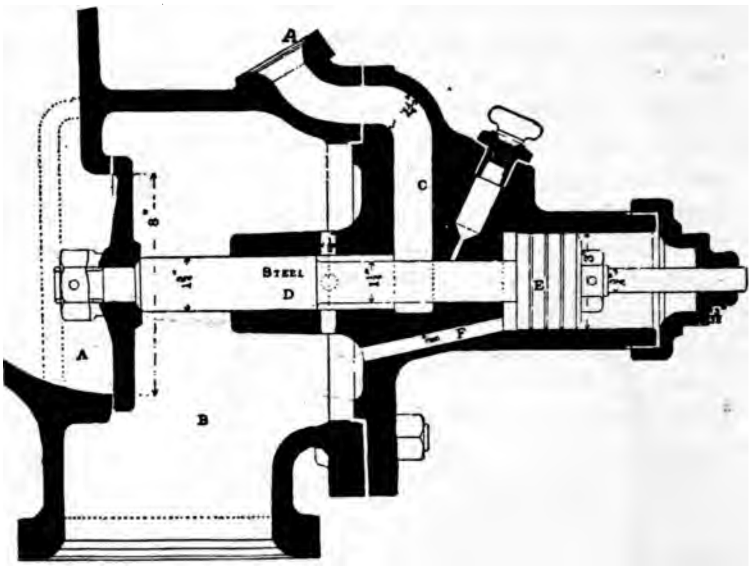


FIG. 58.
Recent Modification of von Borries Intercepting Valve.

When the h. p. cylinder exhausts into the chamber *A*, the pressure, which has heretofore been equal to that of the atmosphere, rises on that side of the valve and thus balances the receiver pressure. Then, as the area of *E* is greater than that of the shoulder on *D* the valve is moved to the right, and the communication between the h. p. exhaust and the receiver is again established. In this position the larger portion of the stem *D* closes the ½ inch openings and the engine works as a compound. It may be added that the openings are so graded that the steam is wire-drawn down to the proper pressure for admission to the l. p. cylinder.

93. Recent Changes in the von Borries System.—After several years careful watching of the locomotives fitted with automatic intercepting valves, Mr. von Borries has reached the conclusion that it is better to give to the engineer a control over the intercepting valve, and to provide a separate exhaust for the h. p. cylinder at starting. With this change in view a new arrangement of starting gear has been devised. It is described, with other non-automatic starting gears, in Chapter XVII, 116.

94. The Worsdell System.—This system is strictly automatic, which means that the change from the use of boiler steam in the l. p. cylinder to full compound action is

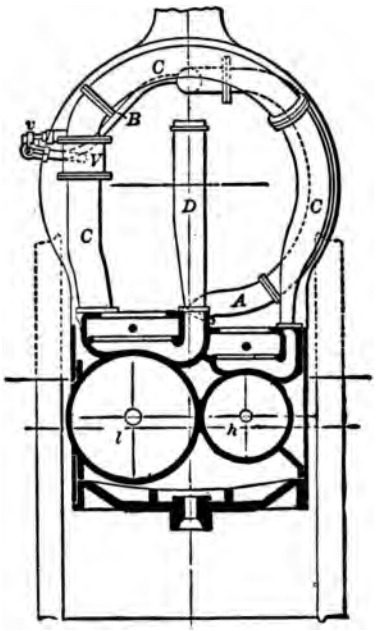


FIG. 59.
Arrangement of Cylinders, Worsdell Two-Cylinder Type.

controlled automatically beyond the will of the engineer, and takes place whenever the pressure in the receiver, resulting from the exhaust of the h. p. cylinder, rises to a point where it is sufficient to actuate the automatic mechanism. In Fig. 59 *h* and *l* represent the h. p. and l. p. cylinders, respectively, *A* is the h. p. steam pipe, *C* is the receiver, *D* is the l. p. exhaust pipe, *B* is the steam

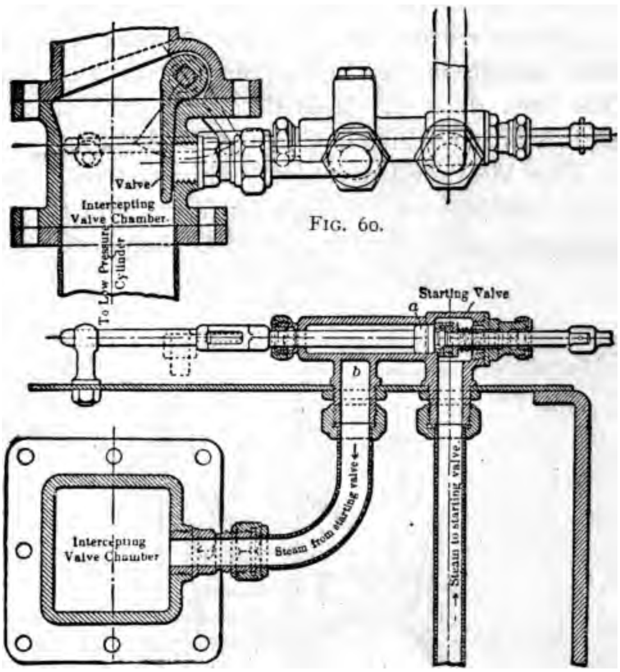


FIG. 60.

FIG. 61.
Early Form of Worsdell Intercepting Valve.

supply to the starting valve *v*, and *V* is the intercepting valve.

The Worsdell starting and intercepting valves are illustrated by Figs. 60 and 61. The intercepting valve is a flap valve, and is shown in Fig. 60 in the position which it occupies when the engine is working as a compound, being swung to one side, and thus leaving a straight, clear passage by it. The spindle on which the valve turns passes out through the side of the smoke box, and carries an arm, which is connected to the small piston shown at *a*, Fig. 61, in a manner which is clearly indicated in the figures. The starting valve casing is connected to the main steam pipe by a small pipe, which is shown in Fig. 61, and also in Fig. 59. The piston *a*, which operates the intercepting valve by means of the connection previously referred to, works in a cylinder which is an extension of the starting valve casing.

A small port, which is covered by a spring-loaded valve, connects this cylinder with the pipe *b*, and thus to the intercepting valve chamber. The starting valve is operated by a lever, and is a double valve, a slight movement of the lever opening the smaller valve, and further motion opening the larger valve, which is then partially balanced.

The operation of these valves in starting is as follows: The starting valve being opened by the engineer, steam, at boiler pressure, acts upon the small piston *a*, and moves it forward or to the left in the Fig. 61. By the same movement the intercepting valve is swung up and closed, and the port connecting with the pipe *b* is uncovered, thus admitting steam from the boiler to the intercepting valve chamber below the valve, and thence to the l. p. steam chest. As the exhaust takes place from the h. p. cylinder, the pressure in the receiver, above the intercepting valve, rises until it is sufficient to open that valve, when, by its movement, the small piston *a* is returned to the position shown in Fig. 61, and the steam supply is thus shut off.

95. **A Modification of the Worsdell System.**—This is shown in Figs. 62 and 63. It is automatic in action, as it allows live steam to be admitted to the l. p. cylinder at starting and automatically cuts off this supply, thus converting the engine into a compound when the receiver pressure has been raised to the proper point by the h. p. exhaust.

When the engine driver opens the throttle valve, steam is admitted through the holes *A A* over the stems of the plungers *C C*. These plungers are then forced to the right, pushing the main valve against its seat and opening the port holes *B B* that connect with the chamber *E*, as shown in the cross section, Fig. 63, which leads directly to the steam-chest of the l. p. cylinder. Thus the live steam is

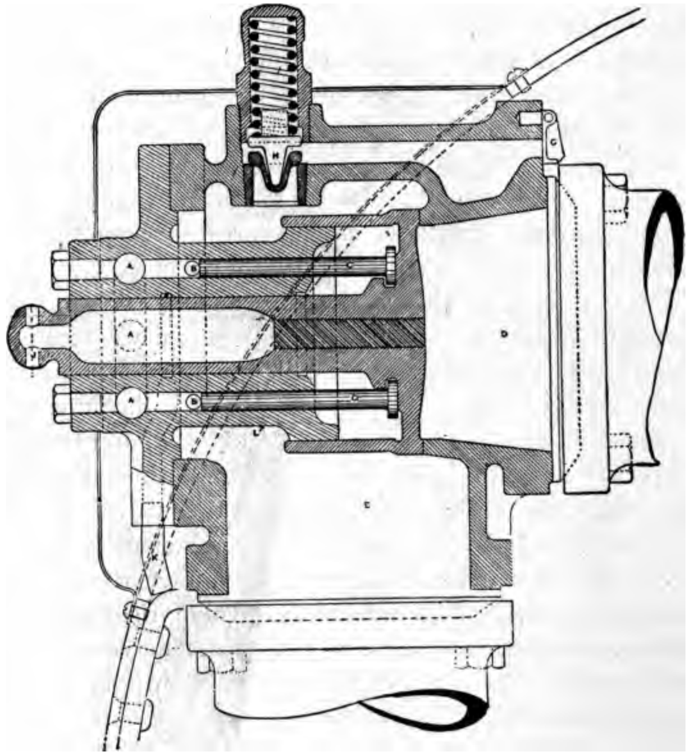


FIG. 62.
Recent Modification of Worsdell Intercepting Valve.

also admitted below the relief valve *H*, so that should the pressure in the l. p. steam-chest rise above that desired, this valve will open and allow the excess of steam to escape into the smoke-box. The small flap valve *G* which closes the passage *F* from the safety valve is used to prevent an accumulation of cinders collecting about the safety valve as a result of long disuse. A drop pipe *K* is also provided to carry off the water condensation and leakage from the annular space *O*.

After the h. p. cylinder has exhausted into the chamber *D*, the pressure in that chamber rises so that finally the

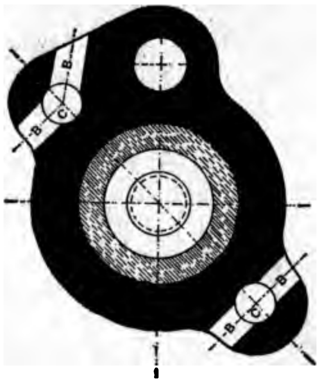


FIG. 63.
Modification of Worsdell Intercepting Valve.

pressure on the under side of the valve overcomes that on the stems *C C*, and the valve opens, re-establishing communication between the exhaust *D* of the h. p. cylinder and the receiver *E* of the l. p. At the same time the stems *C C* close the ports *B B* and the engine proceeds with its work as a compound. The plug shown screwed into the valve is merely used to plug up the core hole made in casting the valve.

96. **The Schenectady Locomotive Works (Pitkin) System.**—This system is strictly automatic, inasmuch as the change from the use of steam directly from the boiler into the l. p. cylinder is controlled automatically and is beyond the will of the engineer. The change from the use of steam directly in the l. p. cylinder to full automatic action occurs whenever the exhaust pressure from the h. p. cylinder accumulates in the receiver to a point where it will actuate the auto-

matic mechanism. The general arrangement of the cylinders and steam connections of this locomotive is show by Fig. 64. The distinctive feature of the engine is the intercepting valve, which is shown by Fig. 65, which is a plan of the bushing which incloses the valve, and by Fig. 66 which is a vertical section through the valve, bushing and saddle.

The valve is shown in the position which it occupies in starting; that is, before compound working begins. In this

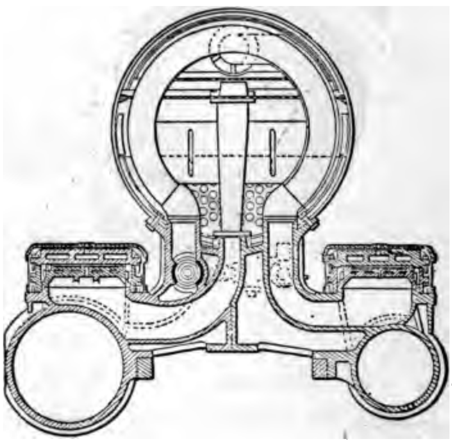


FIG. 64.
Arrangement of Cylinders and Receiver, Schenectady (Pitkin) Type.

position the ports *c* and *d* are closed by the intercepting valve and the connection between the l. p. steam chest and the receiver is thus cut off. The small port *a*, Fig. 65, is connected by a pipe and a pressure-reducing valve to the h. p. steam pipe. By this means steam at reduced pressure is admitted to the space *b* and thence, as indicated by the arrow, to the l. p. steam chest. As the parts of the valve on either side of *b* are of different diameters, the pressure in this space tends to hold the valve in the position shown in Fig. 66. When the locomotive starts, the h. p. cylinder exhausts into the closed receiver, and the back pressure thus created acts upon the forward end of the intercepting valve by means of the passage shown at *e*. The pressure in the receiver rapidly increases until the total pressure on the

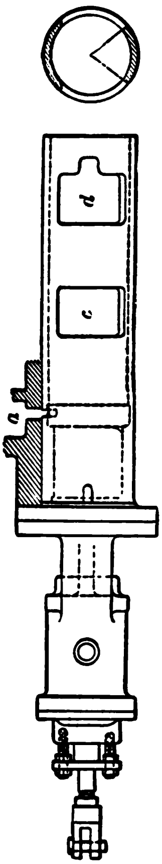


FIG. 65.

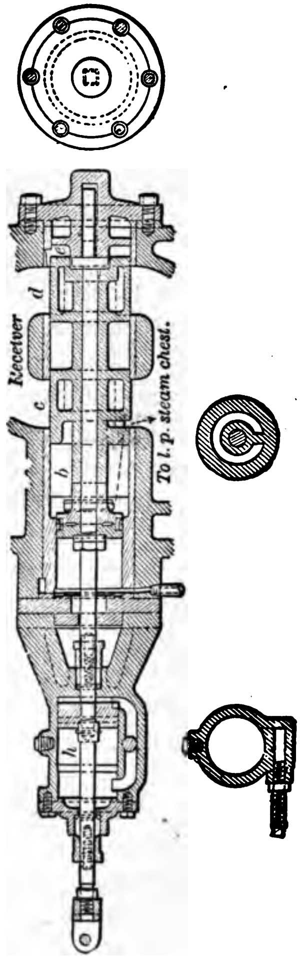


FIG. 66.

Details of Schenectady Automatic Intercepting Valve.

forward end of the valve is sufficient to overcome the total effective pressure at *b*, when the valve is forced to the back end of its stroke, the direct steam supply to the l. p. cylinder is cut off, and compound working begins. To prevent the valve moving too rapidly a dash-pot, in the form of an oil cylinder, *h*, is added. The valve stem is continued through this oil cylinder, and is connected by levers to an index in the cab which indicates the position of the valve.

97. **A Modification of the Schenectady Locomotive Works (Pitkin) System.**—This system is illustrated by Figs. 67 to 71 inclusive. There are two pistons *A A* at one end of the single stem *B*, which moves to and fro in a cylindrical chamber having three openings. Two of these openings, *C C*, lead to the receiver and to the l. p. steam chest, and it is the office of the pistons *A A* to open and close these large openings and prevent the steam in the l. p. steam chest from entering the receiver when it is not wanted there. The other opening, *D*, in this cylinder, connects the intercepting valve cylinder with the l. p. steam chest. There are holes through the pistons *A A* which admit the l. p. steam chest pressure to the right hand end and thus balance these pistons and prevent movement by either receiver pressure or by the pressure in the l. p. steam chest.

The remaining portion of the mechanism is the apparatus for driving and connecting the intercepting valve. It is constructed as follows:

On the end of the stem *B*, which passes through a stuffing box in the end of the intercepting valve chamber, there is a piston *E*, which moves in a small cylinder having ports *F* and *G*, one at each end. These ports lead to a valve seat on which is a plain *D* valve not unlike the ordinary locomotive slide valve. This slide valve is moved to and fro by means of a double piston with a stem between, shown at *J* and *K*. These pistons are of different diameters, *K* being larger than *J*; and as they move to and fro they carry with

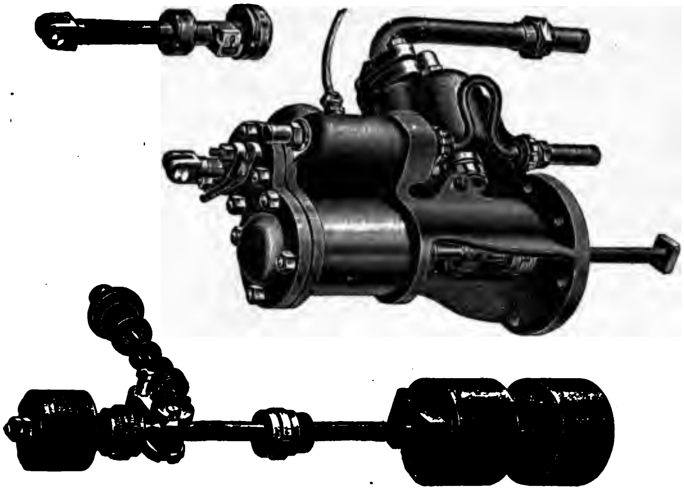


FIG. 67.
Modification of the Schenectady Automatic Intercepting Valve.
Complete Details.

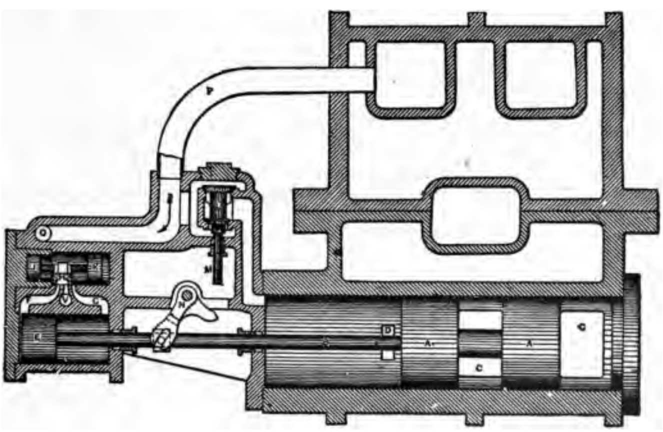


FIG. 68.
Modification of the Schenectady Automatic Intercepting Valve.
Plan Intercepting Valve Open.

them the slide valve. The office of this portion of the mechanism is to move the intercepting valve *A A* to and fro as desired.

The third part of the device consists of a balance poppet valve *L*, which is placed in the path of steam coming direct from the boiler to the l. p. cylinder to assist in starting. This valve has an extended spindle, *M*, on the lower side, and is lifted by means of a bell crank, *N*, which is driven by means of a trunnion on the intercepting valve stem *B*. As the stem *B* passes to the right, the valve *L* is lifted, and as

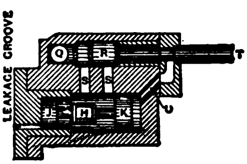


FIG. 69.

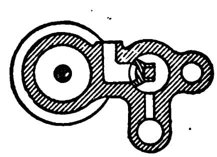


FIG. 70.

Details of Modification of Schenectady Automatic Intercepting Valve.

it passes to the left the valve *L* is allowed to fall. Fig. 69 is a detail of the pipe connections and passages leading to the pistons *J K*, the office of which will be described in what follows. Fig. 70 is a section through the slide valve *H* showing that it has a cylindrical seat. The operation of this valve is a follows:

The engineer opens the throttle, as usual. Boiler steam passes through the pipe *P*, which is tapped into the h. p. steam pipe to the apparatus which actuates the intercepting valve, as shown in Figs. 68 and 69. It enters through *Q* and forces the small regulating valve *R* to the right and then passes down through the left port *S* between the pistons *J* and *K*. *K* being larger than *J*, it has a greater total pressure; hence, the pistons move to the right and carry the slide valve with them. This opens the port *F* and allows the steam to pass on the left side of the piston *E*, and forces it, together with the intercepting valve *A A*, to the right until it is in the position shown in Fig. 71, with the *C C* passages closed. The position of the pistons *J K* and the slide valve *H* at this time are shown in Fig. 71.

During the foregoing operation, as the intercepting valve stem *B* moves to the right it carries with it the bell crank *N* to the position shown in Fig. 71, thus lifting the balance poppet valve *L* and admitting steam, as shown by the arrows, Fig. 71, into the intercepting valve cylinder, from whence

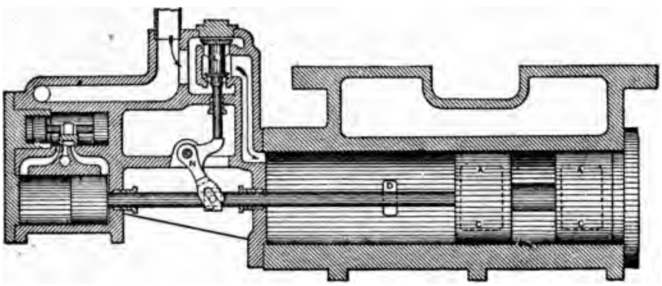


FIG. 71.
Modification of Schenectady Automatic Intercepting Valve.
Intercepting Valve Shut.

it passes out through the opening *D* into the l. p. cylinder steam chest, and in this way steam is admitted direct from the boiler to the l. p. steam chest always just before the engine starts.

As soon as the engine has started and there is an exhaust into the receiver from the h. p. cylinder, steam passes from the receiver through the pipe *T*, shown in Fig. 69, to the passages *U* leading to the piston *K*. This pressure acts on the right hand side of the regulating valve *R*, moves it to the left thus opening the right port *S* and also acting on the larger piston *K*, moves the slide valve *H* and opens the steam passage *G*, Fig. 68, and the exhaust passage *V*, and admits steam to the right hand side of the piston *E*, and drives it to the left, and with it the intercepting valves *A A*, thus opening the passages *C C* and the receiver to the l. p. steam chest. At the same time the bell crank *N* is moved to the left and the valve *L* is allowed to drop into the position shown in Fig. 68, thus cutting off the connection between the boiler and the l. p. steam chest. After this the engine works in the well-known way of the two-cylinder compound; that is, by taking steam into the h. p. cylinder,

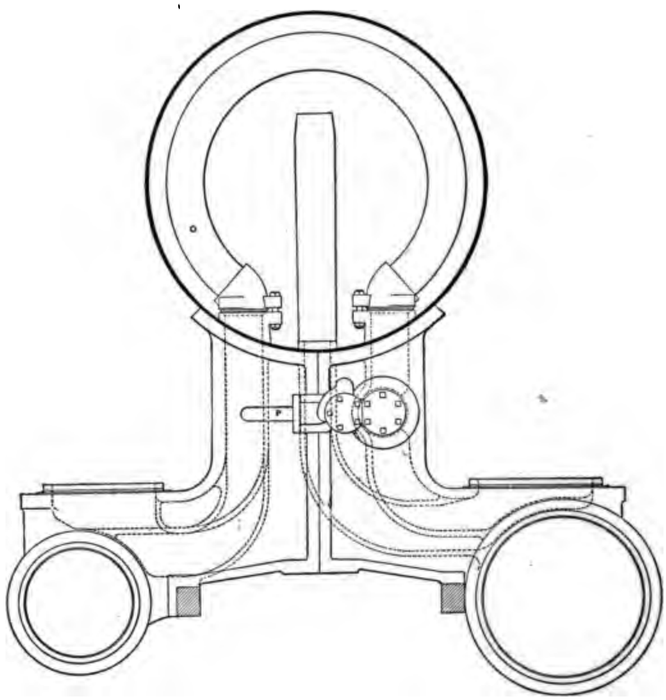


FIG. 71a.
Location of Schenectady Modified Intercepting Valve

discharging it into the receiver, taking it out of the receiver into the l. p. cylinder and discharging it into the atmosphere. Fig. 69 shows the external appearance of the mechanism.

One of the Schenectady two-cylinder compounds on the Southern Pacific has been fitted with an independent exhaust for the h. p. cylinder. The arrangement is simply a piston valve attached to a receiver pipe that is actuated from the cab. At starting, or whenever it is desirable to run the engine with a separate exhaust for the h. p. cylinder, the engineer moves a handle in the cab which opens the piston valve to the atmosphere.

98. **The Dean System.**—This system is strictly automatic, inasmuch as the change from the use of steam directly from the boiler into the l. p. cylinder is controlled automatically, and is beyond the will of the engineer. The change from the use of steam directly in the l. p. cylinder to full automatic action occnrs whenever the exhaust pressure from the h. p. cylinder accumulates in the receiver to a point where it will actuate the automatic mechanism. In the first design the intercepting valve operated almost exactly as that now used, but it was located in the smoke box. The converting valve was placed on the h. p. steam chest cover as shown in Fig. 74. In the present design the intercepting valve and converting valve are joined together and are located on the h. p. steam chest. The receivers are made of cast iron with ribs, as shown in Fig. 72.

99. **A Modification of the Dean System.**—Recently this gear has been modified, and the intercepting and converting valves are bolted to the top of the h. p. steam chest cover, and have connected to them a ½-inch steam pipe for conveying live steam to the intercepting valve for lifting it and securing it in its highest position. See Fig. 72. This pressure is from the boiler and is exerted at all times whether the engine is running or not.

The h. p. main slide valve is open at the top, and the exhaust steam from the h. p. cylinder passes upward through it and a port in the balance plate into the steam chest cover, instead of down through a port in the cylinder as usual; by a passage shown in Figs. 72 and 75 it passes to the receiver.

The starting valves consist of a converting valve and an intercepting valve. The former seats over a hole in the live steam part of the steam chest cover, see Figs. 73 and 74. When the throttle valve opens the converting valve is lifted and steam passes through into the intercepting valve,

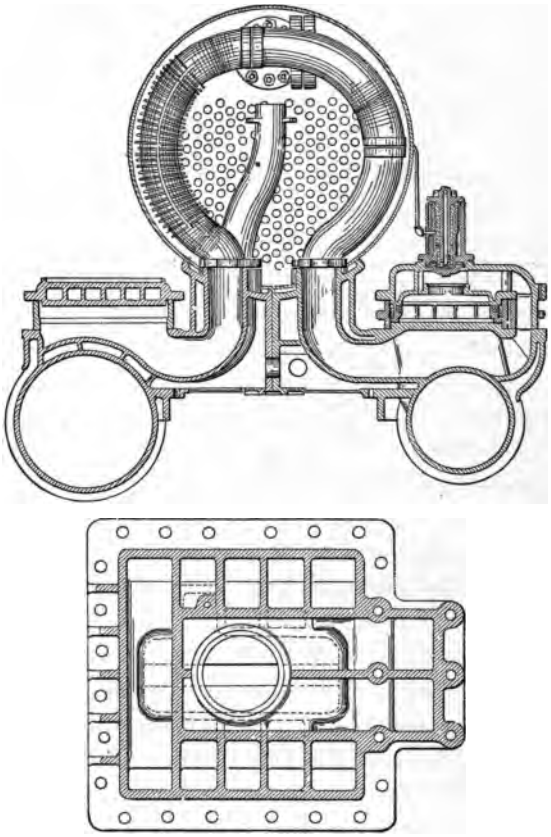


FIG. 72.
Dean's Automatic Intercepting Valve—Cross Section Through Cylinders, and Plan of Steam Chest.

see Fig. 73, which is forced down slowly on account of the steam being wire-drawn through small holes in the intercepting valve, and because of the boiler steam that holds up the intercepting valve, see Fig. 75. When the intercepting valve is nearly on its seat radial holes in the valve allow live steam from the converting valve to pass through into the receiver and into the l. p. cylinder. Thus the l.

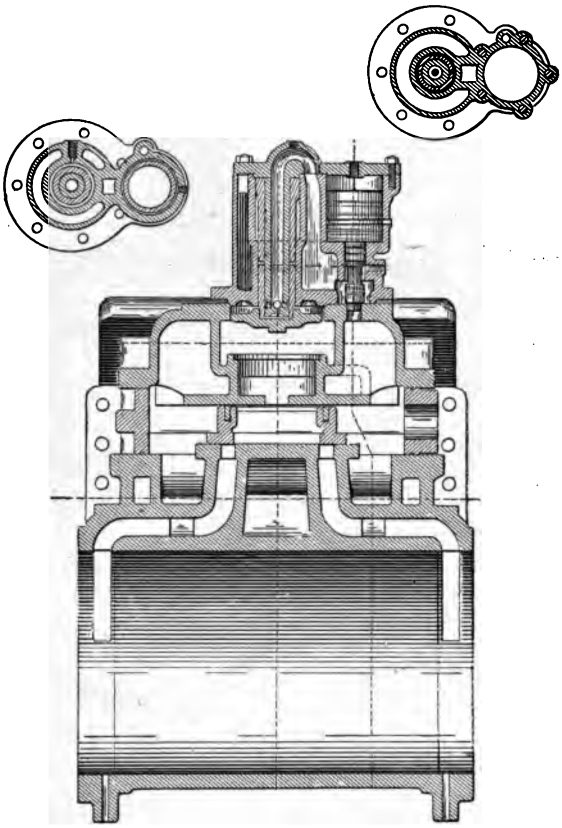


FIG. 73.
Dean's Automatic Intercepting Valve—Longitudinal Section Through High-Pressure Cylinder.

p. cylinder receives steam for starting. When the h. p. cylinder exhausts, the pressure in the receiver acts through a passage, shown in Fig. 74, on the top of the converting valve, moves it downward, and shuts off the supply of live steam. At the same time the grooved stem at the bottom of the converting valve allows the steam that is holding the intercepting valve down to escape into the atmosphere, and thus enables the boiler steam in the annular space around the intercepting valve to lift that valve. The engine then acts as a compound engine.

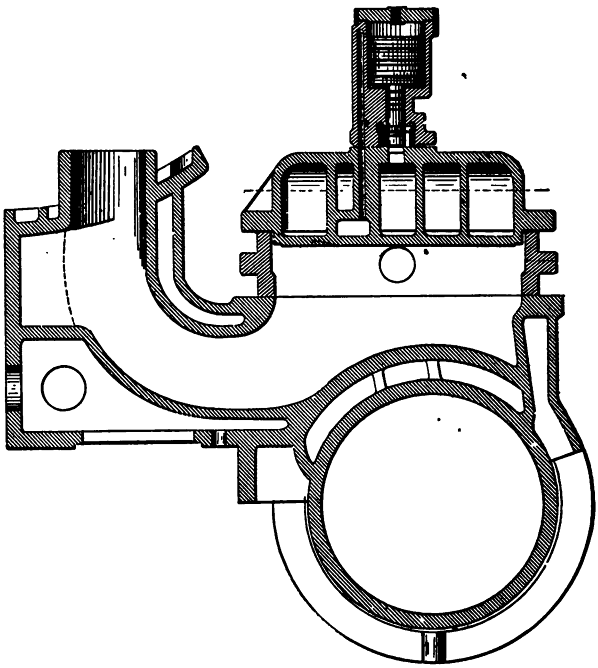


FIG. 74.
Dean's Converting Valve—Cross Section Through High-Pressure Cylinder, before Modification.

In order to prevent the steam, coming from the converting valve at starting, from getting under the intercepting valve, the disc of that valve enters a lip around its seat before the starting steam is allowed to enter the receiver.

Both valves are cushioned to prevent slamming in either direction, and provision is made for oiling.

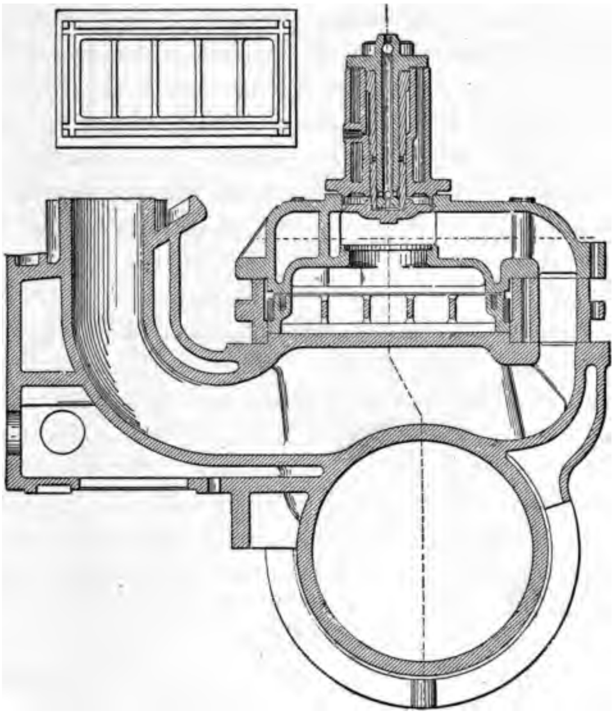


FIG. 75.
Dean's Modified Automatic Intercepting Valve — Cross Section Through High-Pressure Cylinder.

100. The Brooks Locomotive Works (Player) System.—This system is strictly automatic, inasmuch as the change from the use of steam directly from the boiler into the l. p. cylinder is controlled automatically, and is beyond the will of the engineer. The change from the use of steam directly in the l. p. cylinder to full automatic action takes place whenever the exhaust pressure from the h. p. cylinder accumulates in the receiver to a point where it will actuate the automatic mechanism. It is shown in Figs. 76, 77, 78, 78a.

The exhaust from the h. p. cylinder passes into the receiver as usual. Its passage to the l. p. cylinder is governed by an intercepting valve, shown in detail in Figs. 78 and 78a. A pipe leads from the main steam pipe to the end of the intercepting valve, as shown in Figs. 76 and 77, and the steam entering there when the throttle is opened forces the duplex piston forward and closes the intercepting valve, as shown in Fig. 78. The intercepting valve is formed of an annular piston which works on the outside of the duplex piston, as shown. As the duplex piston moves forward, steam is admitted through the interior of that piston, see Fig. 78a, and into the receiver, and passes thence to the l. p. cylinder. In this way the pressure in the receiver increases and finally returns the duplex piston to its seat, see Fig. 78, and stops the admission of boiler steam to the l. p. cylinder. This last movement is caused by the pressure in the receiver acting on the larger piston of the duplex piston, against the steam pipe pressure acting on the smaller piston of the duplex piston. In this way the duplex piston becomes a reducing valve, which reduces pressure of the steam between the steam pipe and the l. p. cylinder according to the area of the two pistons of the duplex piston. When the pressure in the receiver has been raised by the exhaust from the h. p. cylinder, the intercepting valve is forced open and the admission of steam from the steam pipe is shut off by the valve-end of the duplex piston which is forced back to its seat. There is an outlet to the atmosphere which prevents the pressure accumulating on the back side of the annular piston of the intercepting valve, see Figs. 78 and 78a.

In order to move or stop the engine quickly when desired, as for round house work, two valves are attached to the receiver, one on the h. p. side and the other on the l. p. side, which can be opened by a lever in the cab. The opening of the valve on the h. p. side permits the engine to be used like a single expansion engine in a very limited way, as the exhaust from the h. p. is allowed to pass to the atmosphere through this comparatively small auxiliary valve. The valve on the l. p. side of the receiver is kept shut while the engine is being moved, but when it is desired to stop

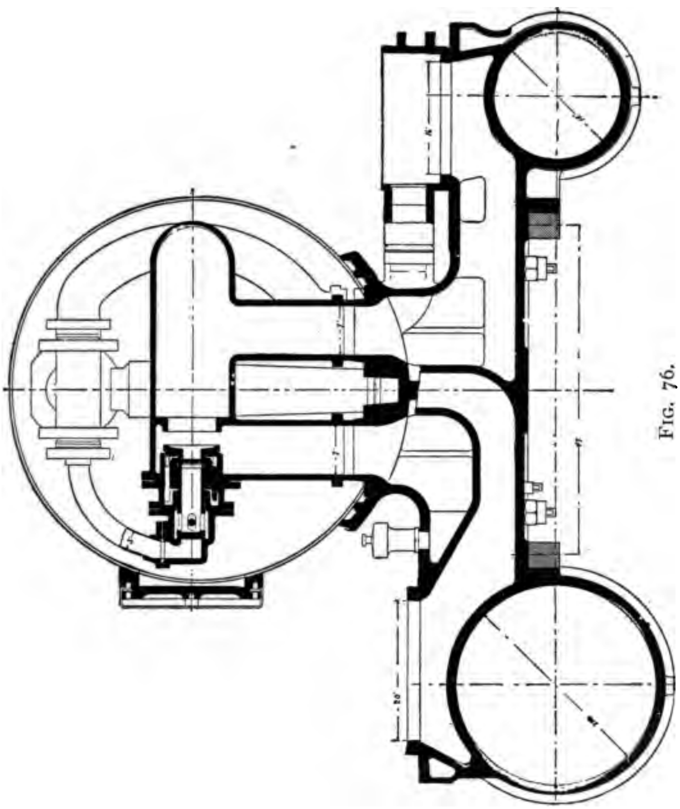


FIG. 76.

Brooks (Player) Automatic Intercepting Valve—Cross Section Through Cylinders. Intercepting Valve Open—Reducing Valve Closed.

quickly, the opening of this valve permits the escape of all the steam in the l. p. steam-chest and in the l. p. side of the receiver. In this way the locomotive is stopped quicker than it would be if the cylinders had been used compound.

101. Rogers Locomotive Works System.—This system is strictly automatic, inasmuch as the change from the use of steam directly from the boiler into the l. p. cylinder is controlled automatically and is beyond the will

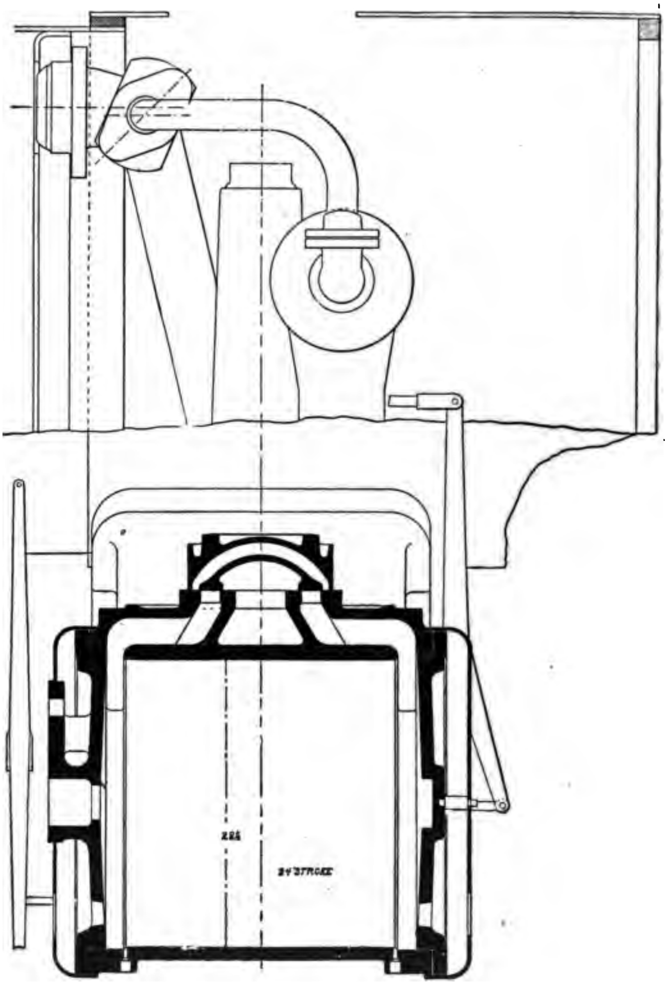


FIG. 77.
Brooks (Player) Automatic Intercepting Valve—Longitudinal Section Through Low-Pressure Cylinder.

of the engineer. The change from the use of steam directly from the l. p. cylinder to full automatic action takes place whenever the exhaust pressure from the h. p. cylinder accu-

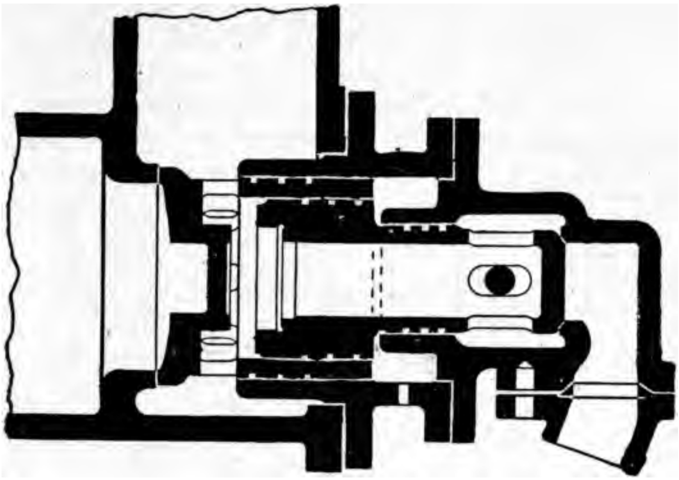


FIG. 78.
Brooks (Player) Automatic Intercepting Valve — Reducing Valve Closed.

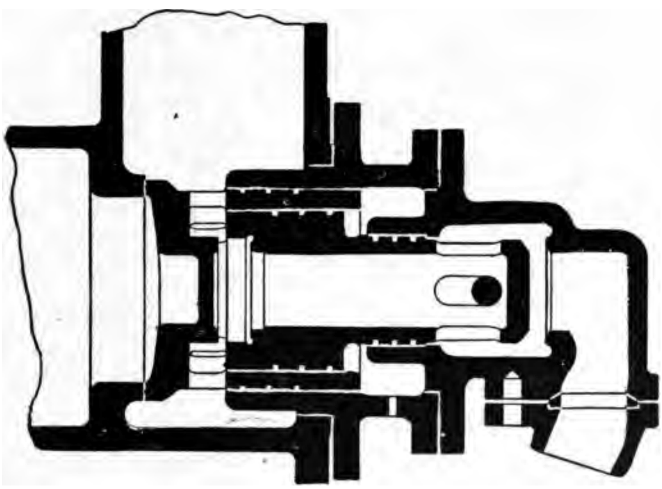


FIG. 78a.
Detail of Brooks Automatic Intercepting Valve—Reducing Valve Open.

mulates in the receiver to a point where it will actuate the automatic mechanism. It is shown in Figs. 79, 80 and 81.

The intercepting and reducing valves are shown in detail in Fig. 79. The reducing valve consists of a valve *B* and piston *A*, mounted on a stem *F*, in an iron chamber *J*, the space between the valve and piston being filled by steam supplied from the live steam pipe through a 2½ inch con-

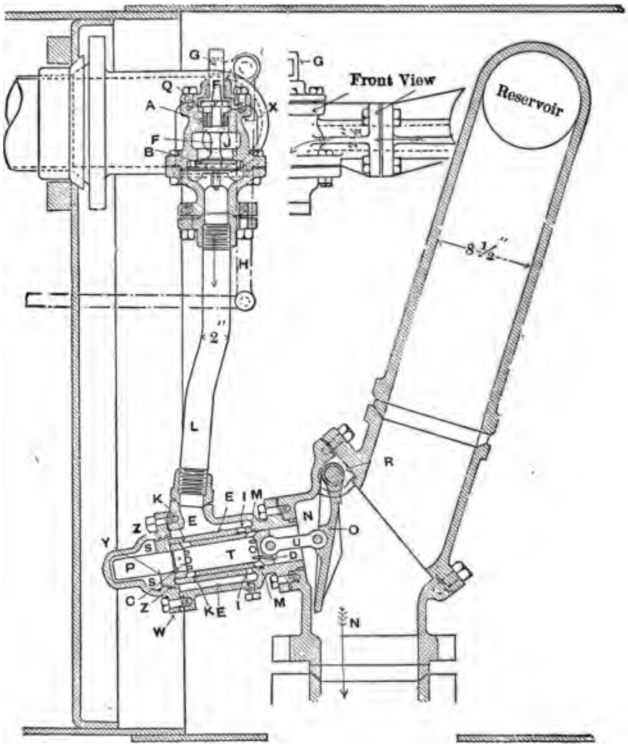


FIG. 79.
Details of Rogers Automatic Reducing and Intercepting Valves.
Intercepting Valve Open.

nection. The net area of the upper side of the valve *B* is 8.30 square inches, while that of the under side of piston *A* is 3.96 square inches. The chamber *Q*, above piston *A*, opens to the atmosphere through port *X*, so that any leakage past the piston will not interfere with the free action of the valve. Neglecting friction, the valve will open when the pressure beneath the valve drops below 52 per cent. of the live steam pressure in the valve chamber *J*, thus admitting live steam to the passage, which leads to the intercepting valve.

The opening of this reducing valve is controlled by the position of the reverse lever, the arrangement being such that the reducing valve can open only when the reverse lever is in the extreme backward or forward gear. Refer-

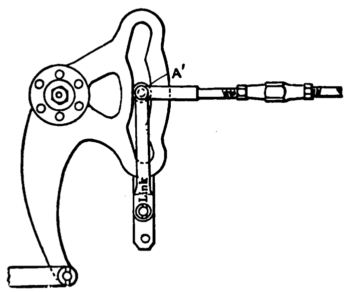


FIG. 80.
Rogers Automatic Intercepting Valve — Details of Cab Connections.

ring to Fig. 79, it will be seen that the upper end of the stem *F* of the reducing valve is slotted to receive the short arm at *G*. This arm is mounted on a short shaft, to which is keyed a longer arm *H*, the end of which drops nearly to the centre of the smoke box. Attached to this arm will be seen a rod leading back to the mechanism, shown in Fig. 80. This device is actuated by an independent reach rod from the reverse lever, Fig. 81. The shape of the curved slot on this mechanism is such that when in mid-gear the arm *G* lifts on the valve stem *F* of the reducing valve with such force as to prevent its opening, but when in extreme forward or backward gear, the tension of this rod is released by the friction wheel *A'*, Fig. 80, passing into the incline of the curved slot at either end, then the arm *G* drops to such a position as to allow the valve to open or remain closed, according to the pressures in and below the rcducing valve.

The steam, after passing through the reducing valve, flows through the 2 inch pipe *L* to the intercepting valve. The valve proper consists of a plain flap valve *O* which closes diagonally across the receiver pipe in such a way as to prevent the steam admitted to the l. p. cylinder from backing up against the h. p. piston and reducing its power. This flap valve is connected to a hollow piston *T* by means of the link *U*. Around the wall of the cylinder in which the hollow piston *T* is loosely fitted is an annular steam chamber *E E* connected with the pipe *L*. Through this cylinder wall there are eight ⅝ inch holes at *I I* and through the wall of the hollow piston *T* there are also eight holes, $\frac{9}{16}$ inches diameter at *K K*, which, when the piston *T* moves outward, correspond with holes *I I*, and steam from *E* will then pass through into *T*. *T* also has eight $\frac{9}{16}$ holes *M M* at its inner end, and as these holes, when the intercepting valve is closed, are outside of the end of the cylinder, steam will pass out through them into the space *N* below the intercepting valve and on to the l. p. steam chest. The head *W*, on the back end of plunger cylinder, Fig. 79, is chambered out as shown at *S S*. In the back end of the hollow piston *T* is a solid plunger *P*. This plunger extends through a hole *Y* in the inner wall of the head into the chamber *S S* fitting loosely in *Y*. From the annular space

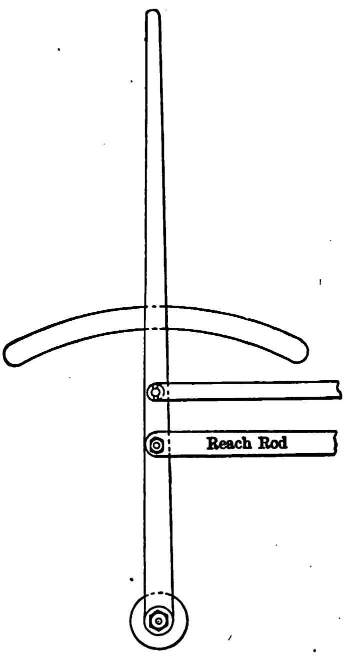


FIG. 81.
Details of Cab Connections—Rogers Intercepting Valve.

E E through the inner wall of the head at *ZZ* are two holes (one top and one bottom) $\frac{1}{16}$ inch diameter into the chamber *S S* for the passage of steam to operate on the plunger *P* in closing the intercepting valve *O*. The dimensions of these parts are as follows:

Diameter of the hollow piston *T* outside, 3 inches.
" " " " inside, $2\frac{1}{4}$ inches.
Stroke to close *O*, about 5 inches.
Diameter of plunger *P*, $1\frac{3}{4}$ inches.

When the parts are in the position shown in Fig. 79, steam is admitted through the pipe *L* to the annular chamber *E*, but as the holes *I I* do not correspond with the holes in the wall of *T*, steam can only pass through the two holes *Z Z* into *S S*, where operating on the end of the plunger *P* it causes the piston *T* to move outward closing the intercepting valve, and at the same time bringing the holes *K K* in correspondence with the holes *I I* and allowing steam from the pipe *L* to pass through the hollow piston *T* out at the holes *M M* at its end into *N* and on to the l. p. steam chest. The object of wire-drawing the steam through the small holes *Z Z* and to have it operate on the comparatively small area of *P* (about 2.4 square inches) in closing the intercepting valve, is to cause as slow a movement of the piston *T* and as light a shock in seating the valve as practicable. There are no steam-tight joints or packing in any of the moving parts for closing the intercepting valve. Whenever the valve *B* of the reducing arrangement is closed, no live steam can get to the l. p. cylinder. To permit the piston *T* to go back.to the position shown, whenever the pressure becomes equal on both sides of the intercepting valve, without resistance, leakage holes are provided at *C* and *D* and by these holes and holes *Z Z* steam can pass through from *N* to *T* to *S* and to *L*. These holes also prevent slight differences in pressure between *N* and *L*, from causing unnecessary movement of the piston *T*.

The locomotives that have been built with this starting gear are given in Table C C, Appendix R.

102. **The Baldwin Locomotive Works System.**—Figs. 82, 83, and 84, show the automatic intercepting valve and starting apparatus devised by the Baldwin Locomotive Works for a two-cylinder receiver compound for the ele-

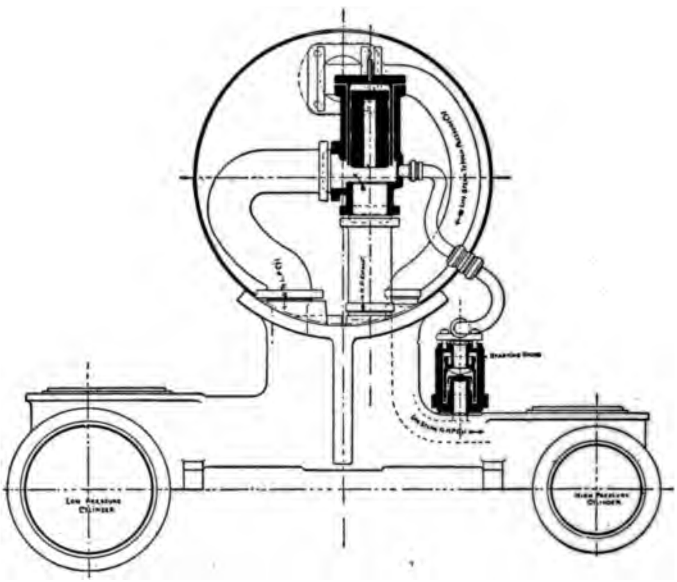


FIG. 82.

Baldwin Automatic Intercepting Valve — Cross Section Through Cylinders.

vated road of the Chicago & South Side Rapid Transit Railroad Company, Chicago. Fig. 82 shows how the steam passes from the boiler to the h. p. cylinder through the steam passage in that cylinder. Opening out of this passage is a starting valve shown in detail in Fig. 84, which is, in fact, a reducing valve, which does not permit the pressure in the receiver to exceed 100 pounds. The boiler pressure is 180 pounds. Whenever there is 180 pounds of steam pressure in the h. p. steam chest and the pressure in the receiver is less than 100 pounds, the reducing valve opens and steam is admitted through a pipe into the receiver. The reducing valve is a single seated valve moved by an annular piston, all of which is cast in one piece, as shown

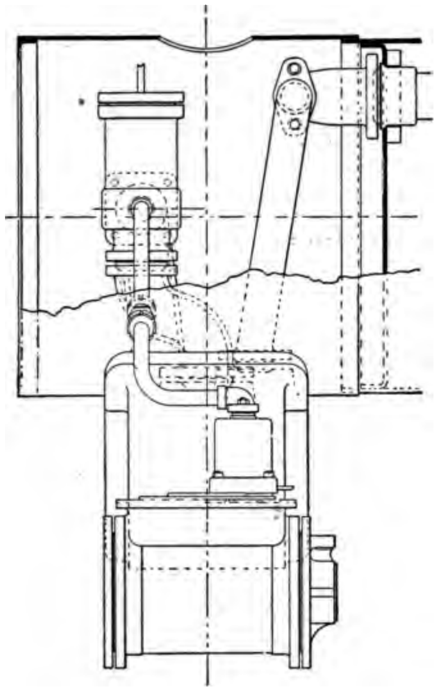


FIG. 83.
Baldwin Automatic Intercepting Valve — Side Elevation.

in Fig. 84. As the piston rises and falls under the variations in steam pressure, the reducing valve is opened and shut.

In the smoke box is an automatic intercepting valve which is opened like other automatic intercepting valves by the exhaust from the h. p. cylinder. This intercepting valve is a simple piston moving vertically in a cylinder formed by the inner casing of a thimble fitted into the receiver. Above the piston there is atmospheric pressure, and below the piston the pressure in the receiver. Hence, the intercepting valve is always open when there is pressure enough in the receiver to lift the valve which is in the form of a plunger. The valve is rather heavy and drops

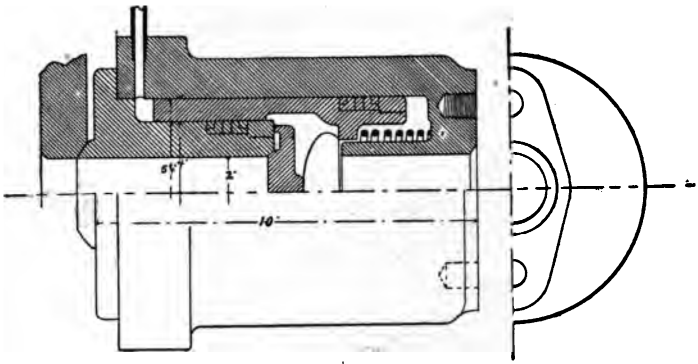


FIG. 84.
Baldwin Automatic Intercepting Valve — Detail of Reducing Valve.

whenever the pressure in the receiver is reduced by the closing of the throttle of the engine. In practical operation the weight and area of this intercepting valve is so arranged that it will keep shut until the engine has made one revolution or less, after which the pressure in the exhaust pipe of the h. p. cylinder has accumulated to an amount that will lift the valve and permit the engine to work compound.

CHAPTER XVI.

DESCRIPTION OF TWO-CYLINDER RECEIVER COMPOUNDS WITH AUTOMATIC STARTING GEAR AND WITHOUT SEPARATE EXHAUST FOR HIGH-PRESSURE CYLINDER AT STARTING AND WITHOUT INTERCEPTING VALVE. THE LINDNER SYSTEM; THE COOKE LOCOMOTIVE WORKS SYSTEM; THE GÖLSDORF (AUSTRIAN) SYSTEM.

103. **The Lindner System.**—This system is not strictly automatic, and perhaps has some advantages for that reason; however, when the engine is operated in the usual way by the locomotive engineer, the system is practically automatic. It is only in the extreme forward and back position of the reverse lever that steam is admitted directly from the boiler to the l. p. cylinder, and as the engines are generally run only for the first two or three revolutions with the reverse lever in the extreme notches, it is evident that under ordinary conditions the engineer would cut out the admission of steam directly from the boiler to the l. p. cylinder by hooking up the reverse lever. If it was desired to use boiler steam in the l. p. cylinder for a longer period, it is only necessary to allow the reverse lever to remain in the extreme notch. The admission valve is shown by Fig. 85. *C* is the receiver, *E* is a small pipe connecting the receiver and the main steam pipe, and *J* is the starting valve, which has two ports, *H* and *I*, formed in it at right angles. The lever *K* by which the valve is operated is connected to the reach rod, and the proportions are such that *K* turns through ninety degrees, as indicated in the figure when the reverse lever is moved from one extreme position to the other. The effect is that steam from the boiler is admitted to the receiver when the valve motion is in either

the extreme forward gear or the extreme backward gear, and the cock is closed for intermediate positions.

Another feature of the Lindner system is the introduction of two small ports, see Fig. 87*a*, of small area in the h. p. slide

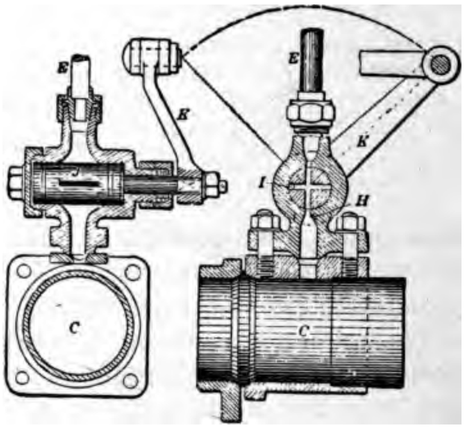


FIG. 85.
Lindner Starting Valve — General Form.

valve, which are so located that when the valve covers the steam port, at one end of the h. p. cylinder, as after cut-off takes place, that end of the cylinder is connected by means of one of these small ports with the exhaust side of the valve and thus with the receiver. The effect is to admit steam at receiver pressure to the end of the h. p. cylinder, which is covered by the slide valve, and as the other end is then open to the exhaust and hence to the receiver pressure, the pressure on the two sides of the h. p. piston is partially equalized. In other words, the effective back pressure on the h. p. piston is more or less reduced, so that it offers less resistance in starting. This device is useful in starting only for such piston positions as lie between full cut-off and the end of the stroke.

The effect of the Lindner starting gear will depend somewhat upon whether or not a relief valve is provided to limit the maximum pressure in the receiver. If this receiver pressure is equal to 1/3 of the boiler pressure, with a cylinder ratio of 2, the effect of the starting valve is to enable the engine to start with very nearly the same distribution of pressures on the pistons as would be found when it is working as a compound in full gear. The resulting rotative efforts will then be represented by a curve such as the full line curve in Fig. 30, the ordinates or actual pressures, however, being less than those for the single expansion engine in about the proportion of 113 to 150, with boiler pressures of 170 and 150 pounds.

If the receiver pressure is allowed to become higher than 1/3 the boiler pressure, the back pressure on the h. p. piston is increased proportionately, and the result is that the power of the h. p. cylinder is reduced, while that of the l. p. cylinder is increased. The advisability of using the higher pressure depends upon the positions of the cranks at starting. If the l. p. crank is at a dead point, the maximum effort will be obtained by not admitting any steam to the receiver at the instant of starting, but before the engine has made 1/8 of a revolution some pressure in the receiver will be necessary to enable the l. p. piston to act. The other extreme is when the h. p. crank is at a dead point in starting. When this is the case, the l. p. crank being then on the half centre, full boiler pressure could be advantageously used in the l. p. cylinder, with the result of obtaining a rotative effort about 4 times as great as in a single expansion engine starting with the same crank positions. But similarly to the first case, the receiver pressure should be reduced almost as soon as the engine begins to move, or else the h. p. piston will be practically thrown out of action, and the engine might be stalled after making 1/4 of a revolution.

It appears, then, that with this starting valve and a properly loaded relief valve on the receiver the starting is very simple; but the power is less than that of the single expansion engine having cylinders of the same size as the h. p. cylinder of the compound, the boiler pressures being 170 and 150 pounds, respectively. With no safety valve, the utility of the device depends upon the position of the crank and the judgment of the engineman, 66.

104. A Modification of the Lindner System.—The latest form of the Lindner system is a modification of the first. It consists of running the pipe which formerly led from the four-way cock to the receiver, into the side of the steam chest. At this point is formed a small valve seat over which rides a flat valve without ports, which is attached rigidly to the valve yoke. This valve is made of such length that when steam is not wanted in the l. p. cylinder for useful effect in starting, it is shut out by the valve, and is not permitted to enter the receiver and back up against the h. p. piston. By it steam can be shut out of the receiver when it is advantageous to do so. It operates in the same general way as an intercepting valve, but has the advantage of being capable of regulation to a greater degree. This is the device used on the present Lindner engines and on the Pennsylvania and C., B. and Q. compounds illustrated in 106 and 107.

A further modification of the Lindner system is sometimes made for locomotives having two h. p. cylinders and one receiver common to both, and for express engines having large driving wheels and comparatively small cylinder power. This modification consists in admitting steam, at starting, to the pipe leading to the l. p. steam chest, through a small auxiliary port in the throttle valve, in such a way that steam is admitted to the h. p. steam chest through the throttle valve in the regular way, before the steam is admitted from the boiler through the small auxiliary port in the throttle valve, to the pipe leading to the l. p. chest. In this way full pressure is admitted to the h. p. steam chest before steam is admitted to the auxiliary pipe leading to the l. p. steam chest. The object of this is to give full pressure to the h. p. piston when that piston has to start the engine before any steam is admitted to the l. p. steam chest or receiver. This arrangement is easily adapted for throttles in the form of a slide valve, but is difficult to apply to engines having a double poppet valve.

105. **The Lindner System as Used on the Saxon State Railroad; The Meyer-Lindner Duplex Compound.**—A four-cylinder compound of the Duplex type has been built for the Saxon State Railroad by the Chemnitz Engine Works with the Lindner starting gear. The engine is known as the Meyer-Lindner Duplex Compound. It is in reality a double two-cylinder compound with receiver, there being two h. p. cylinders on one motor truck, which exhaust into a common receiver which feeds two l. p. cylinders on the other motor truck. The ratio of the cylinders is 2.35. It is claimed for duplex engines of this type, as it is for the Mallet duplex engines, that if the l. p. driving wheels slip they will be stopped at once, because while the slipping is going on the steam required for the l. p. pistons will exceed the amount delivered from the h. p. cylinders, and the turning moment on the driving wheels of the l. p. truck will thus be decreased. In the same way, if any slip should occur with the h. p. truck wheels, it will quickly be stopped, because then more steam would be going from the h. p. cylinders than the l. p. cylinders could receive, and there would be a rapid increase of back pressure from the receiver, with a corresponding decrease of the power exerted on the drivers.

106. **The Lindner System on the Chicago, Burlington & Quincy Railroad.**—Two compound locomotives of the Mogul type have been built by the Chicago, Burlington & Quincy Railroad at the Aurora shops, from designs of Mr. William Forsyth, Mechanical Engineer of that road.

The first engine had the early form of the Lindner gear, that is, without the ports in the sides of the steam chest of the l. p. cylinder over which the fixed valve on the side of the valve yoke passes. This locomotive has given excellent service since it was first built. It has now the

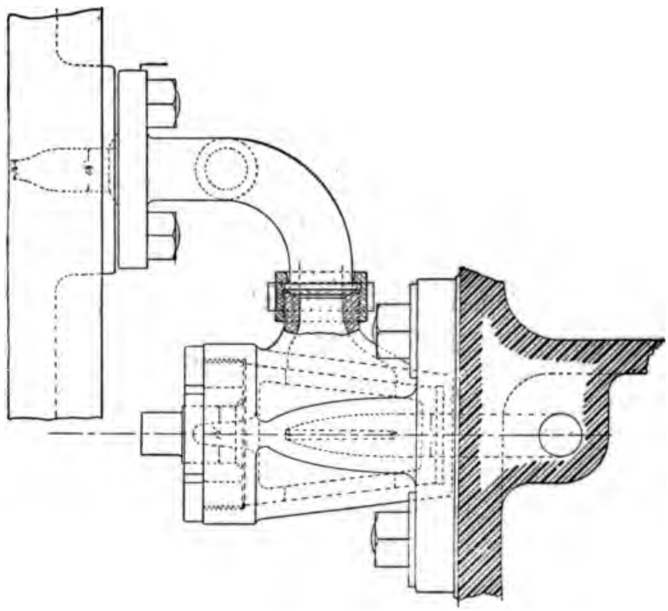


FIG. 86.
Lindner Starting Valve on C., B. & Q. Compound.

latest form of Lindner gear except the connection to the throttle. Figs. 35 and 36 show the general arrangement of cylinders, receiver and the pipe to the starting valve, and Figs. 86, 86*a*, 87 and 87*a* show the valves and their application to the engine in question. Figs. 86, 86*a* and 87 show the application of the starting valve to the l. p. steam chest and cylinder saddle, and also the fixed valve on the valve yoke which keeps the admission port for live steam into the l. p. cylinder closed, except when it is desired that steam should be admitted. Full control of the admission of steam into the l. p. cylinder is obtained in this way. Fig. 87*a*

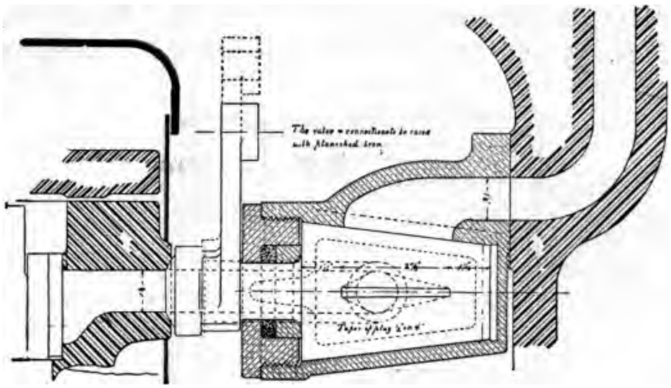


FIG. 86a.
Lindner Starting Valve on C., B. & Q. Compound—Section.

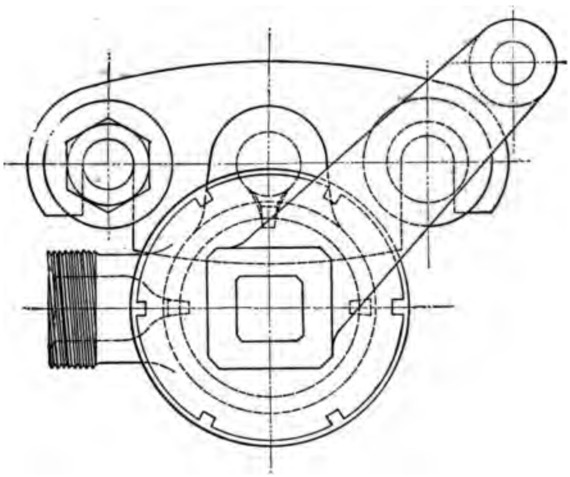


FIG. 87.
Lindner Starting Valve on C., B. & Q. Compound—End View.

shows the h. p. steam valve in section, and indicates how the small balancing ports used with the Lindner system are introduced in the h. p. steam valve. Figs. 35 and 36 give the location of the receiver and the piping for the starting valve. The starting valve is connected by a rod with a supplementary vertical arm on the reverse shaft. This is shown in Figs. 35 and 86*a*. With this arrangement the locomo-

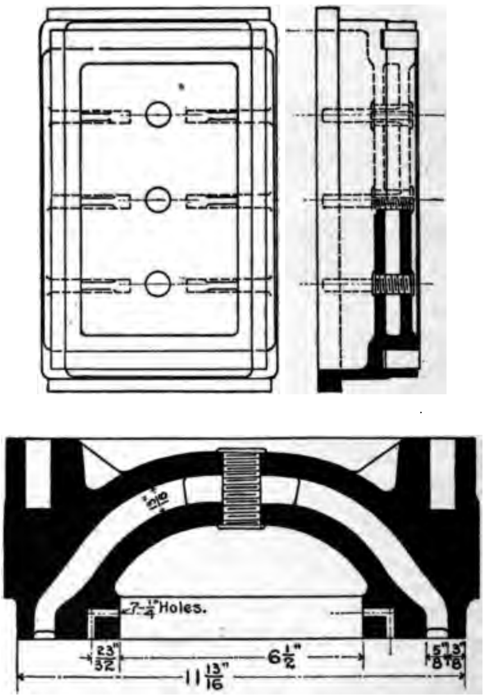


FIG. 87a.

Main Steam Valve of Lindner Compound on C., B. & Q.

tive starts freight trains in regular service, and all ordinary passenger trains, without difficulty. The C., B. & Q. road adopted this arrangement on account of its simplicity. The second engine, with the compound system, is also built with this device.

107. The Lindner System on the Pennsylvania Railroad.—A very interesting compound on the Lindner system has been built by the Pennsylvania Railroad at the Altoona shops from the design of Mr. Axel S. Vogt, Mechanical Engineer. The engine was built for the heaviest class of passenger service, and has been in service but a short time

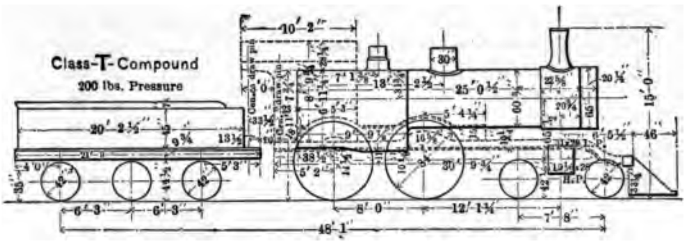


FIG. 88.
Pennsylvania Compound with Lindner Starting Gear—Side Elevation of Engine.

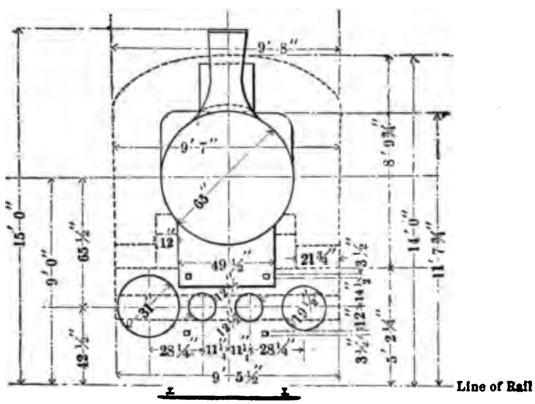


FIG. 89.
Pennsylvania Compound with Lindner Starting Gear—End Elevation of Engine.

at this writing. Some changes were required in the valve motion and smoke box apparatus to improve the steaming of the engine, and she has been taken from service to have these changes made. This is undoubtedly the heaviest four coupled compound locomotive yet made. The general type of the engine is shown in Figs. 88 and 89. The following are the principal dimensions:

Weight of Engine Empty	130,000 Lbs.
" on Drivers	84,000 "
" " Truck	46,000 "
" of Engine in Working Order	145,500 "
" on 1st pair Drivers	48,500 "
" " 2nd " "	46,700 "
" " Truck	50,300 "
Tender Fitted with Scoop.	
Capacity of Tender—Water	3,000 Gals.
" " " —Coal	15,000 Lbs.
Weight " " —Empty	37,100 "
" " " —Loaded	77,000 "
Spread of Cylinders	79 Inches
Distance bet. Centre of Frames	42 "
Width of Cab	9 ft. 7 in.
Height of Cab Roof from Rail (Centre)	14 ft. 0 in.
Inside Length of Fire-Box	9 ft. 0 in.
" Width " "	40 Inches
Number of Tubes	289
Length " "	11 ft. 9½ in.
Outside Diameter of Tubes	1⅞ Inches
Diameter of Drivers, outside of tires	84 Inches
Diameter of Truck Wheels, outside of tires	42 "
Fire Box of Belpaire type.	
Grate area	30 square feet
Fire Box Heating Surface	159 " "
Tube Heating Surface	1661 " "
Total Heating Surface	1820 " "
Working Boiler Pressure, by gauge	200 lbs.
Safety Valve set at	205 "
High-Pressure Cylinders	19½ × 28 in.
Low-Pressure Cylinders	31 × 28 in.

The main valves are of the piston type, and both are 12½ inches in diameter with a maximum travel of 7 inches in full gear, and are placed between the frames in the saddle. The section of the valves is reduced near the centre of the length, and the annular cavity thus formed communicates with the live steam pipe in the h. p. valve and with the receiver in the l. p. valve, so that the steam for both cylinders is admitted at the centre and discharged at the ends of the valves. The steam admission opening to both valves is 5¼ inches wide, and the steam ports leading from valve liner to both cylinders are 2¾ inches wide; making allowance for the bridges crossing the port openings the actual length of the steam ports in the valve seat of both cylinders is 29 inches. The receiver pipe is made of copper and has an internal diameter of 8 inches.

The cut-off in full forward gear is 22 inches or 78.6 per cent. of the stroke in h. p. cylinder, and 23.5 inches or 83.9 per cent. of stroke in l. p. cylinder. The lap on steam side is 1.53 inches on the h. p. valve, and 1.31 inches on the l. p. valve. The clearance or negative lap on exhaust side is 0.625 inch on h. p. valve and 0.75 inch on l. p. valve.

The steam admission leads are as follows:

FULL FORWARD GEAR.

H. p. front, 0.115"
H. p. back, 0.23"
L. p. front, 0.25"
L. p. back, 0.125"

FULL BACK GEAR.

H. p. front, negative 0.70
H. p. back, negative 0.55
L. p. front, negative 0.47
L. p. back, negative 0.70

When in full forward gear the maximum port openings to steam are:

H. p. cylinder, 1.97
L. p. cylinder, 2.19

and to exhaust:

H. p. cylinder, full
L. p. cylinder, full

When cutting off at 50 per cent. stroke the port openings to steam are:

H. p. cylinder, 0.98
L. p. cylinder, 1.12

and to exhaust:

H. p. cylinder, full
L. p. cylinder, full

The radius of the link is 51 inches and length of the connecting rod 7 feet 8 inches. The receiver volume is 26672 cubic inches, the volume of h. p. cylinder 8362.2 cubic inches and its clearance 1045 cubic inches. The volume of the l. p. cylinder is 21134.4 cubic inches and its clearance 1438.7 cubic inches. H. p. cylinder clearance is 12.28 per cent.; l. p. cylinder clearance is 6.8 per cent.; ratio of receiver volume to h. p. cylinder volume is 3.2. The Lindner device, consisting of a four-way plug cock, is applied to this engine, the equalization ports in the h. p. valve are each $\frac{3}{16}$ inches × 1 inch and the controlling port in the l. p. valve is $\frac{7}{16}$ inches × 1½ inches. The pipe which admits steam to the four-way cock is connected directly to the main steam pipe in the smoke box, as has been described before for the Lindner system, 106.

This engine has ⅝-inch inside clearance or negative lap for h. p. cylinders, ¾-inch for l. p. cylinders, and every endeavor has been made to get the best possible steam distribution. The receiver is unusually large, and so far as can be seen at this time the design of cylinder apparatus is one that should give a superior steam distribution, and thus be very economical. Owing to the very liberal clearance, or negative lap, and the large ports used, the cylinder power at high speeds should be greater than any other compound engine locomotive built up to this time. It is intended with this engine to regulate the power with the reverse lever and not with the throttle lever at high speeds.

108. **The Cooke Locomotive Works System.**—An experimental engine was built by the Cooke Locomotive Works, Paterson, N. J., to determine the value of the compound system described in the following: It was a two-cylinder compound with receiver. The cylinders were 19

and 27 × 24 inches. No intercepting valve was employed. The details of the starting gear are shown in Figs. 90 and 91. The volume of the receiver was practically the same as that of the h. p. cylinder. In starting, steam is let into

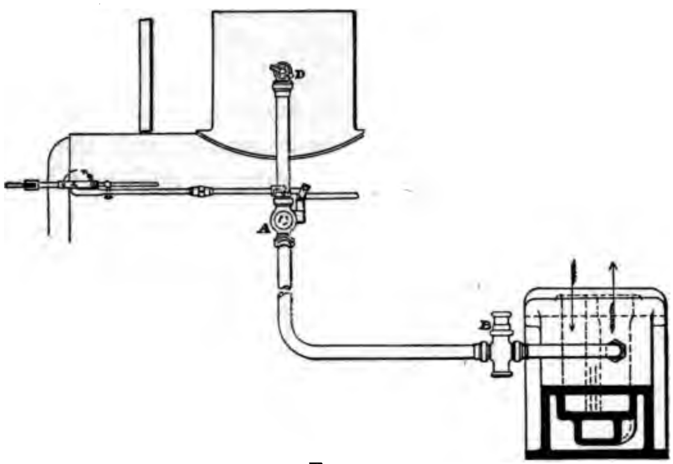


FIG. 90.
Cooke Starting Gear.

the receiver from the dome, by opening a valve *A* which is connected to the throttle lever. Steam passes through a reducing valve *B* and is kept by this valve to the proper pressure. Fig. 91 shows the connection to the throttle lever. When the throttle is closed, the small lever *C* can be operated and the valve *A* opened, but when the throttle is open the valve *C* cannot be opened also, as the lever *C* is then made inoperative by the disengagement of its cam connection with rods leading to the throttle lever.

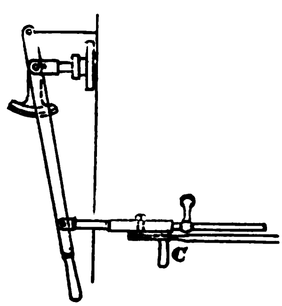


FIG. 91.
Cab Connection, Cooke Gear.

109. **The Gölsdorf (Austrian) System.**—The Austrian Government has made an examination of all the systems of compound locomotives in use. These examinations were made by the mechanical engineers connected with the State Railway system. The reports advised that the increased cost of maintenance of existing types of compounds would be too great under the conditions on Austrian roads, and the matter was dropped for a time; but was taken up again after the invention of a simple starting apparatus by C. Gölsdorf, a mechanical engineer connected with the State Railways. In Austria the coal is inferior, and the Government reports state that but 3½ pounds of water are evaporated per pound of coal used. This coal being inferior

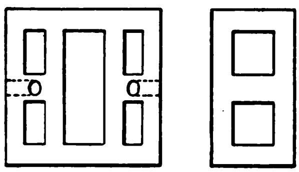


FIG. 92. FIG. 93.
Gölsdorf Starting Gear — Plan of Valve Seat and Valve.

and expensive, the advantage of compounding is somewhat greater in Austria than in other parts of Europe. The saving by compounding was found to be about 18 per cent. Gölsdorf's device is constructed as follows: Leading from the main steam pipe is a 1-inch copper steam pipe which connects with a fitting on the l. p. steam chest, at which the current of steam is divided into ¾-inch pipes which lead to two ports constructed in bridges in the main steam port, as shown in Fig. 92. These ports are about ¾ inches long by ⅝ inches wide in the direction of the valve travel. The steam valve, Fig. 93, has a bridge across its centre, as shown, which covers the small steam ports. This describes the entire construction of the compound starting gear which is in the l. p. valve seat. The h. p. valve seat is constructed as usual. The valve motion is the Walscheart, which has been chosen because it gives a longer maximum cut-off than the ordinary link motion. By it is obtained a maximum cut-off of 92 per cent. The operation of this system is as follows :

When the reverse lever is in full gear, or nearly so, the valve travel is such as to uncover the small port whenever the l. p. cylinder is to furnish the power for starting, and in this way steam enters from the main steam pipe, when the throttle is open, to the l. p. cylinder steam chest and receiver. When the start is to be made by the h. p. cylinder, the l. p. slide valve is in such position as to cover the small port and prevent the entrance of steam from the steam pipe into the l. p. cylinder. When the engine is started, the driver hooks up the reverse lever, which reduces the valve travel and the small ports are not uncovered. With this gear, which is not unlike the Lindner, the maximum starting power can only be obtained during the first revolution, or, more correctly, during a part of the first revolution. The first of these engines was built in 1892. Since then five others have been ordered. A general description of the engines is given in Table C C, Appendix R.

CHAPTER XVII.

DESCRIPTION OF TWO-CYLINDER RECEIVER COMPOUNDS WITH INTERCEPTING VALVE AND WITH SEPARATE EXHAUST FOR HIGH-PRESSURE CYLINDERS AT STARTING.

110. **The Mallet System.**—This system is non-automatic, by which is meant that the change from the use of h. p. steam in the l. p. cylinder to full compound action is made at the will of the engineer and not automatically. This system has suitable valves so that the engine may be operated as a single expansion engine, not only in starting but at any time when in service. Such an engine, while having all the advantages of compound working, possesses an emergency power equal, or possibly superior to, a single expansion engine having the same general dimensions.

Figs. 94 to 99, inclusive, illustrate the arrangement of this system as applied to a converted six-coupled engine of the Western Switzerland Railroad.

In Fig. 94, *h* and *l* are the h. p. and l. p. cylinders, respectively. *A* is the main steam pipe from the boiler to the h. p. cylinder, *B* is the receiver, *C* is the l. p. exhaust pipe, *D* is the starting valve which is connected to the boiler by the pipe *E*, *F* is the intercepting valve, and *G* is the exhaust pipe from the h. p. cylinder when working as a single expansion engine.

The construction of the starting valve is shown in Figs. 95 and 96. It consists primarily of a short slide valve *a*, which, as shown, covers two ports leading to the receiver. The pipe *p* connects the starting valve chamber with the main steam pipe. On the back of the valve *a* is an inverted slide valve *b*, which slides on a seat formed in the valve-chest cover. A small pipe *c* connects the starting valve chamber with the intercepting valve on the other side of the smoke box, as shown at *c*, Fig. 97. Referring now to

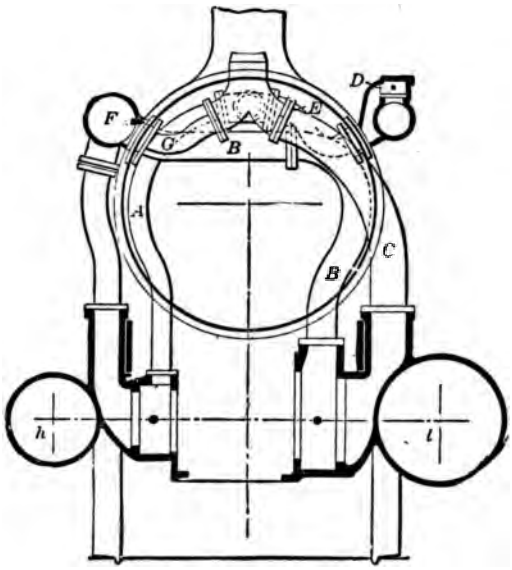


FIG. 94.
Mallet Starting Gear — Arrangement of Parts.

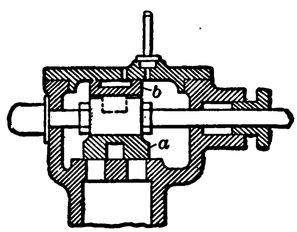


FIG. 95.
Mallet Starting Gear — Detail of Starting Valve.

Fig. 97 it will be seen that the intercepting valve consists of two circular valves and a piston, all being mounted on one stem, and so forming a sort of balanced double poppet valve. The connections to the intercepting valve are as indicated in the figure, the central opening connecting with the h. p. exhaust, the left with the common exhaust nozzle and the right with the receiver pipe.

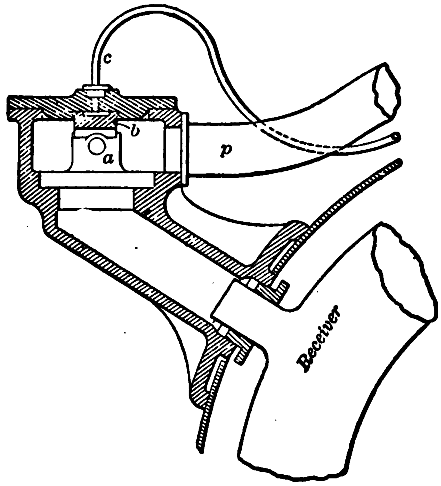


FIG. 96.
Mallet Starting Gear — Detail of Starting Valve.

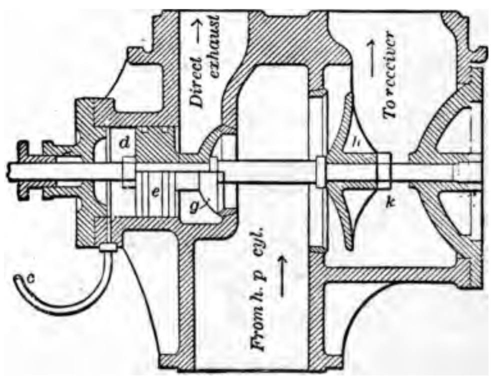


FIG. 97.
Mallet Intercepting Valve.

The operation of these valves is as follows: They are shown in the illustrations in the positions which they ordinarily occupy, or when the engine is working as a compound. Under these circumstances steam from the boiler is admitted to the space *d* back of the piston *e* by way of the small pipe *c*, the starting valve chamber, and the pipe *p*. The pressure thus acting upon the piston *e* keeps the valve *g* closed against the ordinary receiver pressure. The intercepting valve can, of course, be connected so as to be worked by hand in connection with the starting valve. If now the starting valve is opened, or moved to the right in Fig. 95, steam from the boiler is thereby admitted to the receiver, and at the same time the pipe *c* is placed in communication with the atmosphere by means of the cavity in the top of the starting valve. The pressure back of the piston *e* being thus reduced, the valve *g* is opened by the receiver pressure, and the valve *h* is closed, in which position it is retained by the excess of the pressure in the receiver, Fig. 97, or that on the l. p. side of the valve, over that on the h. p. side which is now in communication with the exhaust nozzle. It will be seen that the locomotive will now work as a single expansion engine, and will continue to do so as long as the starting valve is kept open. As soon as it is closed the intercepting valve will be returned to the position shown in Fig. 97.

On the engine illustrated by Fig. 94, a pressure-reducing valve is inserted between the starting valve and the receiver. This reducing valve is of the common differential piston type, adjusted by springs. In addition to this the receiver is fitted with a spring safety valve loaded to 70 pounds pressure. It would seem when a starting valve of this form is used in conjunction with a safety valve, that the introduction of a reducing valve is unnecessary, as the receiver pressure can be regulated by the starting valve.

111. The Early Form of the Mallet System.—In earlier designs Mr. Mallet has combined the starting and intercepting valve in one distributing valve. This is illustrated by Figs. 98 and 99. The distributing valve and a reducing valve are enclosed in a casing which is fastened to the smoke box. The main steam pipe is connected at *a*, and thence by a passage *b*, back of the valves, to the h. p. steam chest. An opening at *c* admits steam from this pipe to the reducing valve chamber and thence to the distribut-

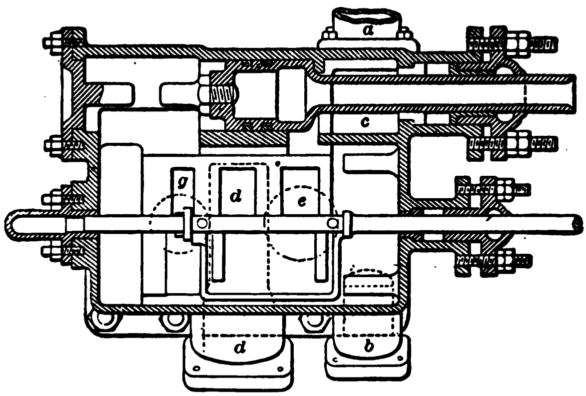


FIG. 98.
Mallet Distributing Valve.

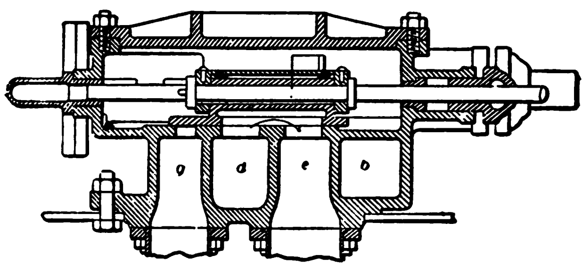


FIG. 99.
Mallet Distributing Valve.

ing valve chamber. The distributing valve is a slide valve, and covers three ports, as shown. Of these *d* is the h. p. exhaust, *e* connects with the receiver, and hence with the l. p. steam chest, and *g* leads to the exhaust nozzle. The valve is shown in the position for compound working. If it is moved forward, or to the left in the illustrations, the passage *d* is connected with *g*, and the h. p. cylinder exhausts directly to the exhaust nozzle, and at the same time by means of the passages *c* and *e* boiler steam at reduced pressure is admitted to the receiver and the l. p steam chest.

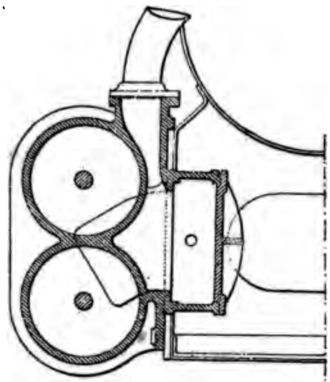


FIG. 100.
Mallet's Proposed Double Low-Pressure Cylinder.

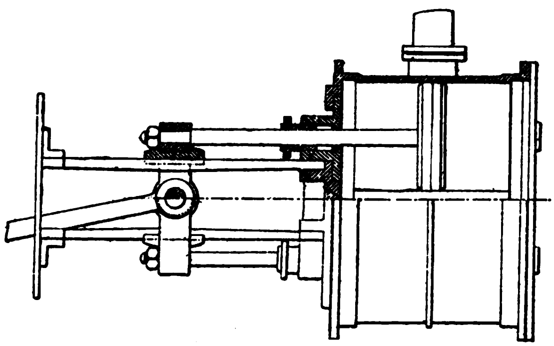


FIG. 101.
Mallet's Proposed Double Low-Pressure Cylinder.

The earlier forms of intercepting valves were not wholly automatic in their action, but required to be closed by hand before opening the throttle in starting. In this form there were no small plungers, and the steam was admitted around the valve stem *k*, which was fluted for part of its length for this purpose. The valve was also connected by a bell-crank arrangement to a weighted arm, which held the valve open and prevented rattling when running with steam shut off.

112. **Preliminary Work of Mallet.**—The earliest work of real practical value in compound locomotive designing was done by Mr. Mallet. Two of his most important contributions to the subject are the separate exhaust of the h. p. cylinder at starting, previously described, and the double l. p. cylinder, Figs. 100 and 101. The object of this double l. p. cylinder is to give to the nominally two-cylinder type the necessary volume of l. p. cylinder without exceeding the maximum width allowable for locomotives.

113. **Rhode Island Locomotive Works (Batchellor) System.**—The Rhode Island Locomotive Works, or Batchellor, system is shown in Figs. 102, 103 and 104. The following is the construction and operation: Fig. 102 shows the front section of intercepting valve at ports *d* and *e*, also front view of portion of receiver with exhaust valve. Fig. 103 shows side section of intercepting valve while running compound. Fig. 104 shows side section of intercepting valve when engine is operating with independent exhaust for h. p. cylinder. *A* is the intercepting valve casing, *B* is the reducing valve, *C* the oil dash-pot, *D* is a pipe from main steam pipe to intercepting valve, *E* is the receiver, *F* is the exhaust valve leading to atmosphere from receiver, *a*, *b* and *c* is the intercepting valve piston, *d* is a port leading from *D* to through the casing of the intercepting valve, *D* being the pipe from the main steam pipe to intercepting valve, *e* is a port from intercepting valve casing to the reducing valve *B*. There is a port from the intercepting valve casing into the passage leading to the l. p. steam chest. *m* is the crank which operates the exhaust valve leading from the receiver to the atmosphere, *o* and *o* are ports leading through the exhaust valve *F* and its seat.

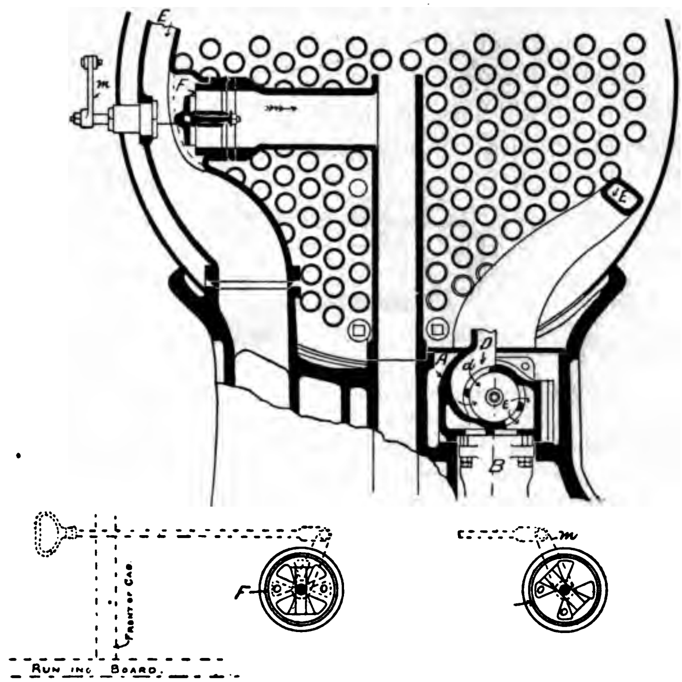


FIG. 102.

Rhode Island Locomotive Works (Batchellor) Starting Gear — Cross Section Through Intercepting Valve and Separate Exhaust Valves.

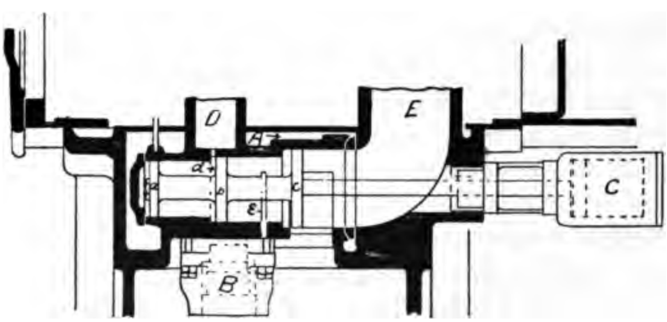


FIG. 103.

Longitudinal Section Through Rhode Island Locomotive Works (Batchellor) Intercepting Valve — Valve Open.

The operation of the device is as follows: The intercepting valve being in any position, as in Fig. 103, and the exhaust valve closed, the throttle being opened, boiler steam will pass to the h. p. cylinder in the usual manner, and also through pipe *D* into the intercepting

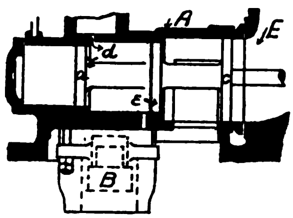


FIG. 104.
Longitudinal Section Through Rhode Island Locomotive Works (Batchellor) Intercepting Valve — Valve Closed.

valve *A*, causing the piston to move into the position shown in Fig. 104. In this position the receiver is closed to the l. p. cylinder by the piston *C*, and steam from *D* passes through ports *d* and *e*, and reducing valve *B*, into the l. p. steam-chest; the pressure being reduced from boiler pressure in the ratios of the cylinder areas. The piston *a-b-c*, is so proportioned that it will automatically change to the compound position shown in Fig. 103, when a predetermined pressure in the receiver *E* has been reached by exhausts from the h. p. cylinder. The engine thus starts with steam in both cylinders, and automatically changes to compound at a desired receiver pressure.

The engine may be changed from the compound system to the single expansion at any time, at the will of the engineer, by opening the valve *F* connecting the receiver to the exhaust pipe, allowing the exhausts from the h. p. cylinder to escape through the nozzle in the usual manner.

The exhaust valve *F* is operated as follows: The lever *m*, which rotates the exhaust valve *F*, is connected by a rod to a handle in the cab. To run compound place lever *m* as shown on the left in Fig. 102, which closes ports *o*. To run single expansion place lever *m* as shown on the right in Fig. 102, the ports *o* opening *E* to exhaust.

It is obvious that, in case of bad conditions of starting, the engine may be operated single expansion at the will of the engineer by opening the exhaust valve before starting, and that upon its closure the piston *a-b-c* will automatically take the compound position of Fig. 103.

This system can be used either as automatic or non-automatic as desired.

The Rhode Island Locomotive Works claim the following for their system of starting gear:

(1) Compound engine automatically adapted to all requirements of variable service.

(2) All necessary devices by which a locomotive may be run at any time and at any place on the road, and for any length of time demanded by the service, as a single expansion engine; each cylinder doing exactly halt the work, whatever that may be, and without waste of steam.

(3) The engineer, at any time he chooses, may change the engine into compound working, permitting it to operate thus as long as circumstances will require, and then he may change it back again at once into single expansion working. These changes are made as easily as the engineer turns his hand to open or close one valve, by a convenient lever in the cab, and can be done when the engine is standing or in motion.

(4) Great simplicity in form and number of working parts, and whose steam-ways are most uniform in section and most direct in course from boiler to point of application.

(5) Ability to run as a single expansion locomotive in case of break down with no more trouble than an ordinary locomotive.

The use of an independent exhaust for the h. p. cylinder has made these engines well adapted for elevated railroad service. This company has built on this plan a number of engines that are in successful service, see Table CC, Appendix R.

114. **The Richmond Locomotive Works (Mellin) System.**—This system is strictly automatic under ordinary conditions; that is, the use of steam directly from the boiler into the l. p. cylinder, is shut off whenever the exhaust pressure from the h. p. cylinder accumulates in the receiver

to a point where it will actuate the automatic mechanism. But it also has what the inventor calls an "emergency" valve, and by it the engineer can open a separate exhaust for the h. p. cylinder for a sufficient period at starting to get the train under way. At this writing the patents for this device have not been granted, and it has been deemed inadvisable to publish the drawings. The following is, however, a general description:

In the cylinder saddle there is a small piston with a dash-pot connected to the piston rod which controls an intercepting valve placed horizontally. The intercepting valve shuts off the steam, that is admitted to the l. p. cylinder at starting, from entering the receiver. Surrounding the small piston just mentioned is an annular sleeve or piston which serves as a reducing valve. The emergency valve consists of a plain, bevel-seated valve attached to a piston which is connected on one side to a live steam pipe leading to a valve in the cab. This piston is returned to its seat by a spring on the piston rod. The device operates as follows:

Steam from the main steam pipe acts upon the annular piston around the intercepting valve stem and forces the intercepting valve to its seat, thus closing communication between the l. p. cylinder and the receiver. At the same time the sleeve or annular piston opens a small port which admits steam to the l. p. cylinder directly from the main steam pipe. This sleeve then acts as a reducing valve. The intercepting valve is prevented from slamming by the air dash-pot on the end of the stem. When the intercepting valve is closed the exhaust from the h. p. cylinder accumulates in the receiver, and pushes the intercepting valve back to an open position and the engine works compound.

When it is desired to work with a separate exhaust for the h. p. cylinder the engineer opens a valve in the cab and admits steam back of the piston, which is connected with the emergency valve. The pressure on the piston forces the emergency valve open. This opens communication from the receiver to the atmosphere, and gives a separate exhaust for the h. p. cylinder. When running, the engine can be changed from compound to non-compound by opening the valve in the cab. This system then, can be used as either automatic or non-automatic, as desired.

115. **The Pittsburgh Locomotive Works (Colvin) System.**—Figs 105 to 107 show the non-automatic intercepting and reducing valve used by the Pittsburgh Loco-

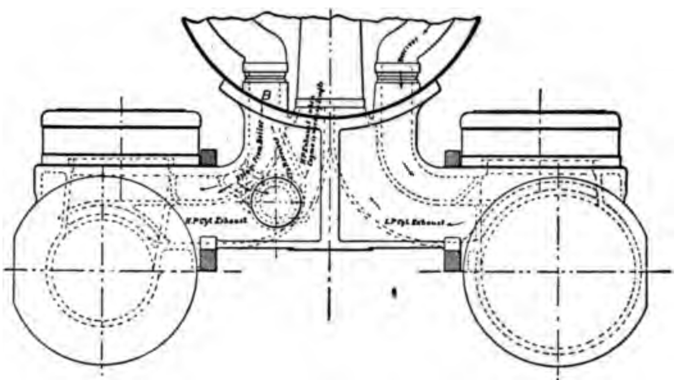


FIG. 105.
Arrangement of Cylinders and Intercepting Valve, Pittsburgh Locomotive Works (Colvin) System.

motive Works on several two-cylinder compounds which they have built. This reducing non-automatic intercepting valve is placed in the h. p. cylinder saddle, as shown in Fig. 105, and is so arranged that the engineer, by moving the lever in the cab, can open an independent exhaust for the h. p. cylinder through passage Fig. 106, to the stack. When it is desired to run compound the lever is again moved and the intercepting valve is open. In Fig. 107 the intercepting and reducing valve are shown when in the position to work compound.

In this system steam from the steam pipe in the h. p. cylinder saddle passes to the reducing valve through a small passage shown in Figs. 106 and 107. When the reducing valve is permitted to open, as it is in Fig. 106 by the removal of the intercepting valve to the right, steam passes directly through the reducing valve as shown by the

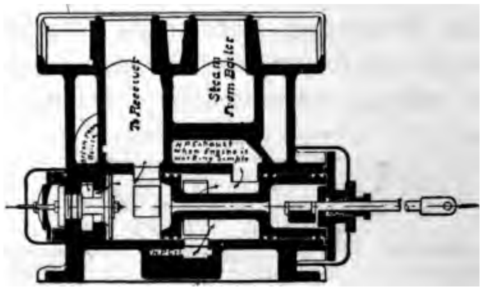


FIG. 106.

Pittsburgh Locomotive Works System—Separate Exhaust for High-Pressure Cylinder, Open.

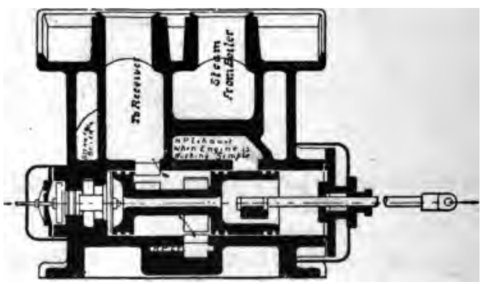


FIG. 107.

Pittsburgh Locomotive Works—Separate Exhaust for High-Pressure Cylinder, Closed.

arrows from the h. p. steam pipe to the receiver thence to l. p. cylinder. The amount of reduction of pressure by the reducing valve depends upon the ratio of the areas of the piston of the reducing valve and the area of the valve itself.

When the engine is to be run compound the engineer forces the intercepting valve back to the position shown in Fig. 107 by means of a rod which is connected to a lever in the cab. The movement of the intercepting valve to the left forces the reducing valve to its seat as shown in Fig. 107 and permits the h. p. cylinder to exhaust into the receiver. When in the non-compound position, shown in Fig 106, the h. p. cylinder exhausts directly to the atmosphere as indicated in Fig. 105.

The engines that have been built with this gear up to this time are given in Table C C, Appendix R.

116. von Borries' Latest System.—After a number of years' experience with automatic starting gears that give increased power to compound locomotives during a part of the first revolution, Mr. von Borries has reached the important conclusion that an independent exhaust with an h. p. cylinder, such as used by Mallet, is necessary for two-cylinder receiver compounds with cranks at right angles when the locomotive has to start heavy trains or work on comparatively heavy grades. Mr. von Borries' device for accomplishing this is as follows:

A double piston valve having a piston rod with a reduced section, which serves as a reducing valve, operates horizontally in a chamber on top of the h. p. steam chest. The chamber has three main passages, one leading to the receiver, one leading to the h. p. exhaust, and a third leading to the atmosphere. This last is the independent exhaust for the h. p. cylinder. This chamber also has a passage connected with a comparatively small pipe leading to the h. p. steam pipe. Through this passage comes the steam that goes directly to the receiver and l. p. cylinder at starting. When the piston is at one end of the stroke the exhaust passage from the h. p. cylinder to the atmosphere is open. When in the other extreme position, the separate exhaust is closed and the passage is open through which the h. p. cylinder exhausts into the receiver. The movement of the piston is accomplished by a steam pressure which is admitted at one end of the double piston through a small valve that is actuated by a lever from the cab. The steam enters the small valve from the h. p. steam pipe through a small copper pipe connecting the two. The piston is cushioned at each end of the stroke by the steam that is being used, and no dash-pots are necessary. At starting the engineer moves a lever in the cab which admits steam back of the piston and closes the intercepting valve and opens the exhaust from the h. p. cylinder to the atmosphere for as long a period as may be desired at starting.

The reducing valve is a part of the stem of the double piston, thus no separate piece is used for it. Drawings are not obtainable at this writing on account of patent complications.

CHAPTER XVIII.

DESCRIPTION OF FOUR-CYLINDER NON-RECEIVER COMPOUNDS, "CONTINUOUS" EXPANSION OR WOOLF TYPE. VAUCLAIN AND NON-RECEIVER TANDEM TYPES.

117. **The Dunbar System.**—A four-cylinder compound locomotive was built by the Boston & Albany Railroad Company in 1883, under the Dunbar patents. The cylinders were 12 inches and 20 inches in diameter, by 26 inches stroke, and were arranged tandem with the h. p. and l. p. pistons on the same piston rod. The engine could be worked compound or non-compound at will. After working about seven months the locomotive was changed to a single expansion engine as it was apparently no more economical than the single expansion locomotives. It is stated that the ports were too small and that the inventor was absent during the trial. As the locomotive was an experiment, it is not surprising under the circumstances that the results were unsatisfactory.

118. **The Du Bousquet (Woolf) System on the Northern Railway of France.**—A successful application of the tandem form of compound engine to a locomotive has been made by Mr. G. Du Bousquet, of the Northern Railway of France. This locomotive is an eight-coupled outside connected engine, all of the weight being on the driving wheels. It was originally a single expansion locomotive, having cylinders 19.68 inches in diameter by 25.59 inches stroke. The boiler pressure of 142.2 pounds, gauge, is the same as before converting it. The principal dimensions of this locomotive are as follows :

Diameter of high-pressure cylinders	15 inches.
" " low-pressure "	26 "
Stroke of pistons	25.6 "
Diameter of driving wheels	51.2 "
Total weight, all on driving wheels	113,970 pounds
Area of grate	22.4 sq. ft.
Total heating surface	1,356 " "

The changes in the distribution and amount of the weights on the axles on account of converting are given as follows :

	Simple.	Compound.
First axle	26,900	29,670
Second axle	24,470	31,390
Third axle	26,670	30,820
Fourth axle	20,500	22,090
Total	98,540	113,970

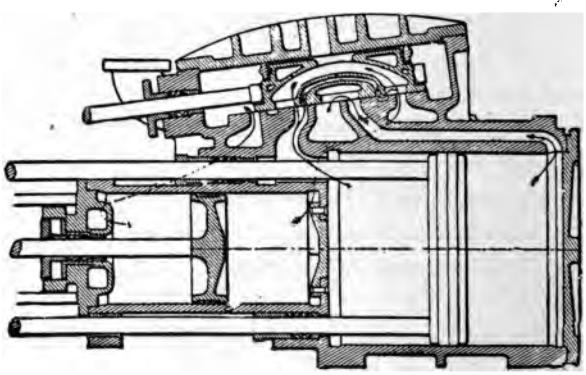


FIG. 108.

Cylinders and Steam Chest of the Du Bousquet Type.

To balance the increased weight of the cylinders a foot board weighing 6,600 pounds was put in. Fig. 108 illustrates the arrangement of the cylinders and valve chest, and is worthy of careful examination. It will be seen that the steam distribution for both cylinders is controlled by one valve, the l. p. valve being, as it were, inside of the h. p. valve. The arrows clearly indicate the paths of the steam. The principal dimensions relating to this valve gear are as follows:

Travel of valve....................................		6.22 ins.
Steam lap, both cylinders, front......................		1.34 "
" " " " back......................		1.22 "
Exhaust lap, high-pressure		0.00 "
" " low pressure		0.32 "
Ports, high-pressure steam	17.72 ins. ×	1.38 "
" low-pressure "	17.72 " ×	1.97 "
" " " exhaust.........................	17.72 " ×	3.54 "
Angular advance of eccentrics		30 deg.
Clearance, per cent., of cylinder volume h. p...........		15.4
" " " " " " l. p...........		7.0
Volume of connecting passages, per cent. of h. p. volume		16.5

The features of this design which are specially noteworthy are that the dead space between the cylinders is reduced to a minimum, the h. p. clearance space is large, and that there are no bushings between the cylinders, but instead there are outside stuffing boxes which are easily accessible.

119. **Indicator Cards from the Du Bousquet (Woolf) Compound.**— The indicator cards shown by Figs. 109 to 113, inclusive, illustrate the steam distribution in this loco-

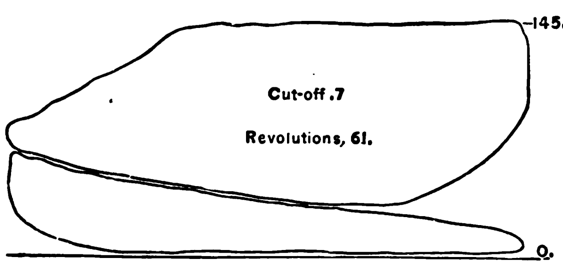


FIG. 109.
Indicator Card at Slow Speed, from Du Bousquet Type.

motive. The effect of piston speed upon the distribution is well illustrated by Figs. 110 and 112, which were taken at the same nominal point of cut-off, but as the two pairs of cards are apparently from opposite ends of the cylinders, it is probable that the great increase in compression shown in Fig. 112 is partially due to irregularity in the valve motion. The mean pressures in these diagrams and the percentage of the total work done in the h. p. cylinder are as follows:

	Mean pressure. H. p.	L. p.	Per cent. of work done in h. p.
Fig. 109	79.36	30.87	46.2
" 110	63.01	21.76	49.1
" 111	51.20	15.36	52.6
" 112	36.84	15.22	44.7
" 113	31.86	9.53	52.7

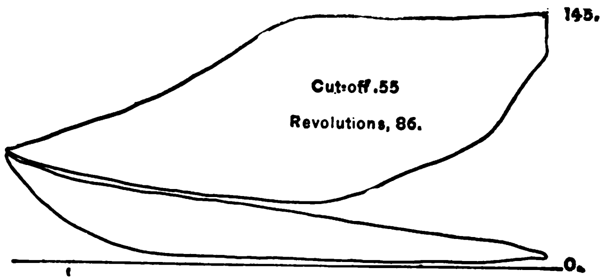


FIG. 110.
Indicator Card, $\frac{55}{100}$ Cut-Off, Du Bousquet Type.

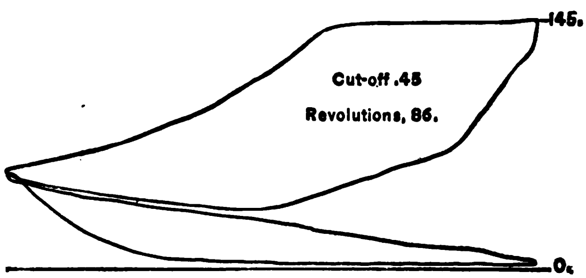


FIG. 111.
Indicator Card, $\frac{45}{100}$ Cut-Off, Du Bousquet Type.

This locomotive has been carefully tested in comparison with a single expansion locomotive belonging to the same original class. The compound hauled trains about 12 per cent. heavier than the single expansion locomotive, with a noticeable saving in fuel, while with trains of the same weight the saving in fuel, as reported by Mr. Du Bousquet, was from 13.5 to 25.8 per cent. The average of five tests is 21.9 per cent.

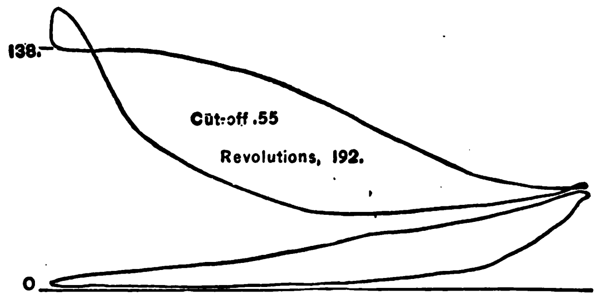


FIG. 112.
Indicator Card, $\frac{55}{100}$ Cut-Off, Medium Speed, Du Bousquet Type.

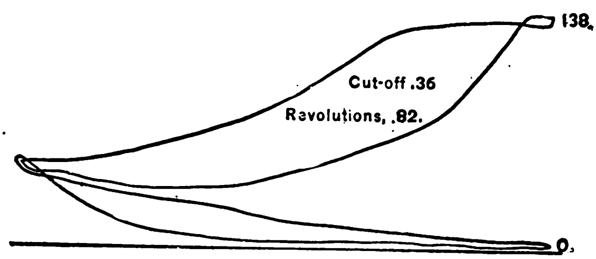


FIG. 113.
Indicator Card, $\frac{86}{100}$ Cut-Off, Slow Speed, Du Bousquet Type.

120. Baldwin Locomotive Works (Vauclain) System.—The first locomotive of this type was built by the Baldwin Locomotive Works in the fall of 1889, and was put to work on the Philadelphia Division of the Baltimore & Ohio Railroad. The general arrangement of the cylinders and valve is shown by Figs. 114 to 123. The method by which the power from both cylinders is transmitted through one crosshead is shown in Figs. 118 and 119, which also shows the direct connections of the valve. The steam distributing

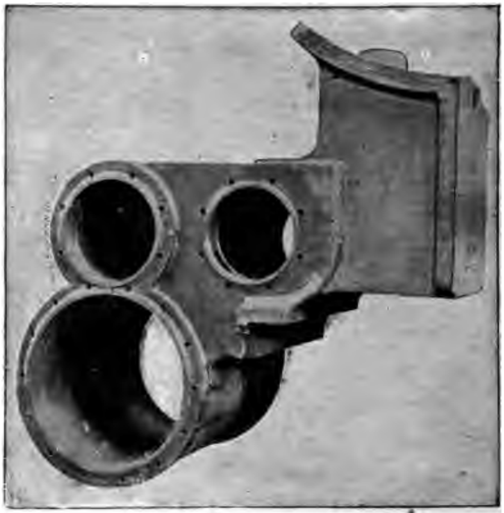


FIG. 114.
Vauclain Cylinders with High-Pressure Above.

FIG. 115
Vauclain Cylinders with Low-Pressure Above.

valve is a hollow piston valve, the action of which is illustrated by Fig. 120.

The cylinders are arranged two on each side, with the l. p. cylinder directly above or below the h. p. cylinder, depending upon the service and the clearances and conditions

FIG. 116.
Vauclain Piston Valve.

to be met. The main valve chamber, which replaces the steam chest of the ordinary h. p. locomotive, is cast in one

FIG. 117.
Vauclain Piston Valve Bushing.

piece with the cylinder casting and is placed as near the cylinders as possible in order to give short steam passages.

The by-pass, or starting valve, is located below the cylinders and main valve. This starting valve is not connected in any way with the valve gear of the locomotive, and is operated from the cab by a small lever located near the reverse lever.

In order to illustrate more clearly the passage of steam through the steam valve to and from the cylinders, the main valve is shown in Fig. 120 as being between the cylinders. For the same reason the starting valve is shown between the cylinders, Figs. 121, 122 and 123.

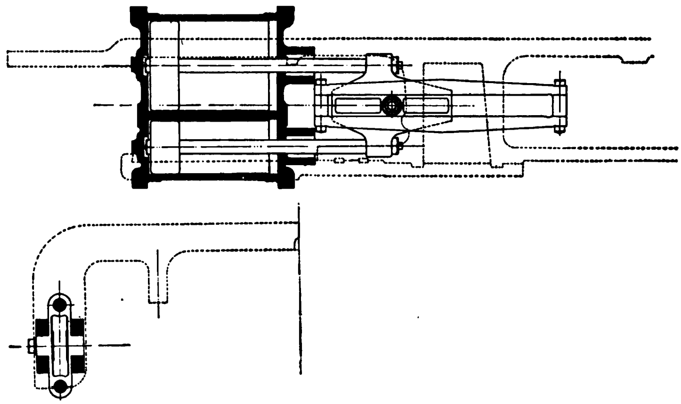


FIG. 118.
Arrangement of Crosshead, Guides and Piston, Vauclain Type.

In this design, at the present time, the air valve, shown in Fig. 116, on the end of the piston valve, is no longer used.

From Fig. 120 it is seen that the steam valve, shown between the cylinders, is a hollow piston with solid ends. A cavity extends around the middle of the valve. The passages and ports lettered *A* are connected directly with the steam pipes leading from the boiler to the valve chamber; those lettered *B* are ports and passages leading from the steam valve to the h. p. cylinder, and those lettered *D* connect the steam valve with the l. p. cylinder. *C* is the final exhaust passage to the atmosphere.

With the valve, as shown in Fig. 120, the steam, at boiler pressure, is entering the valve chamber at the port *A* on the left of the figure, and as the end of the valve does not cover the port *B* on the left, the steam passes from *A* to *B* and so into the front end of the h. p. cylinder, where it expands during the time the port is closed by the valve.

At this time in the back end of the h. p. cylinder there is steam that has been used in expansion and is ready for exhausting into the l. p. cylinder. It now passes through

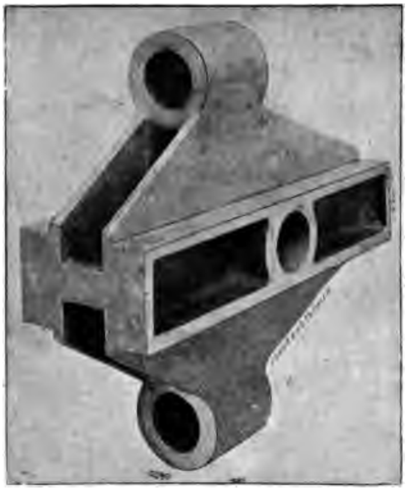


FIG. 119.
Vauclain Crosshead.

the passage *B* on the right, to the steam valve and to the inside of the valve, where it passes from the back end to the front end of the valve into the passage *D* on the left of the figure, and thence into the front end of the l. p. cylinder, as shown by the arrows.

In the back end of the l. p. cylinder is steam that is ready for exhausting into the stack. It has been used in the l. p. cylinder. It now passes from the back end of the l. p. cylinder through the passage *D*, on the right, to the cavity around the steam valve, thence to the exhaust passage and to the atmosphere, as shown by the arrows.

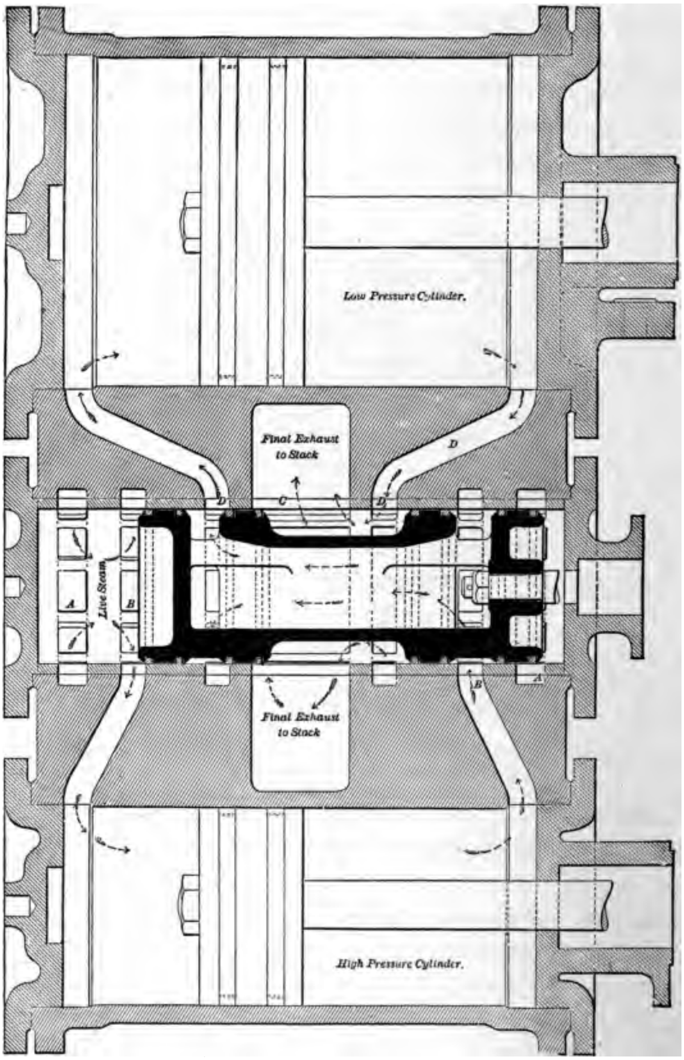


FIG. 120.
Steam Distribution, Vauclain Type.

The simultaneous action of the steam in both ends of both cylinders is as follows: While steam is entering the front end of the h. p. cylinder direct from the boiler past the end of the steam valve, the steam in the back end is exhausting through the steam valve to the front end of the l. p. cylinder, and the steam from the back end of the l. p. cylinder is exhausting into the cavity around the valve, and thence to the exhaust pipe and the atmosphere. This is the course of the steam in the cylinders when the engine is working compound, and, with the exception of the small jet of high pressure steam admitted through the by-pass valve, the same course is followed by the steam when the engine is working in what is called "high pressure."

To make as plain as possible the course of the steam when the engine is working with some of the high pressure steam in the l. p. cylinder, reference is made to Fig. 120. Suppose a pipe to connect the passages *B B*, and to have a valve in it; now, if the valve is open, as it is when the lever in the cab is in its middle or front position, steam can pass freely through the valve and pipe from one passage *B* to the other *B* and balance the h. p. piston. Now this is exactly what takes place when the by-pass valve is used, and it is done as follows:

Steam passes from the boiler into the steam valve chamber, and continues on into the steam passages of the h. p. cylinder. A large part of this steam continues on to the l. p. cylinder, just as when the engine works compound, but the remainder of the steam passes through the pipe and starting valve to the back steam passage *B* on the right of Fig. 120, mingling with the steam that is exhausting from the back end of the h. p. cylinder, and thence to the front end of the l. p., thus increasing the pressure of steam on the l. p. cylinder. This increase goes on until the engine starts. After the engine starts the pistons move so rapidly that the small opening in the by-pass valve cannot supply steam fast enough to keep up the pressure. If the engine does not start readily the pressure in the l. p. cylinder goes on increasing until it reaches boiler pressure. It will be seen that back pressure in the h. p. cylinder is increased by

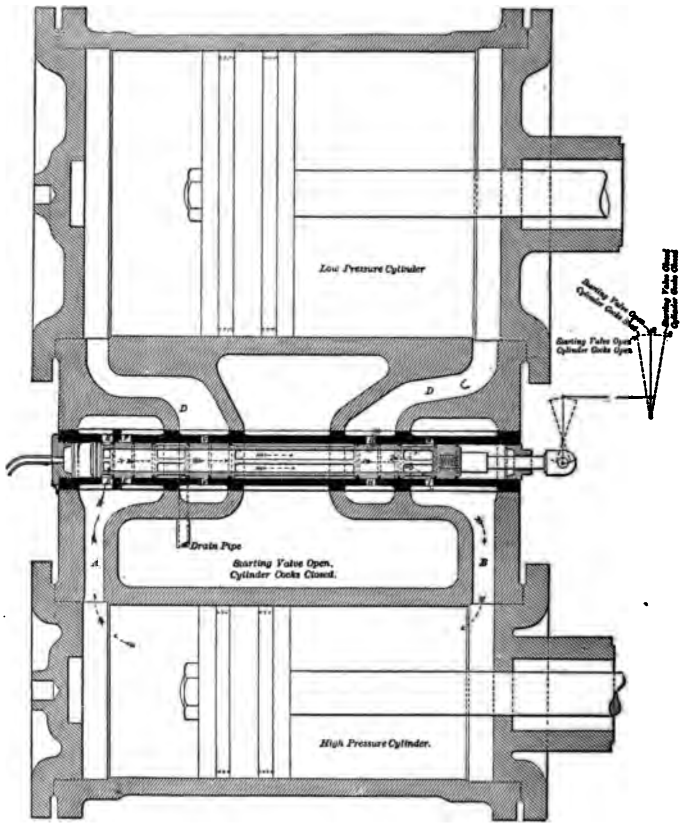


FIG. 121.
Starting Valve and Cylinder Cocks, Vauclain Type.
Starting Valve Open and Cylinder Cocks Closed.

this, and therefore the work done in the h. p. cylinder is less, under these conditions, than when the engine is working compound, but the work done in the l. p. cylinder is much greater than when working compound. The piston in the l. p. cylinder being of greater area than that in the h. p. cylinder, the combined effort of the two

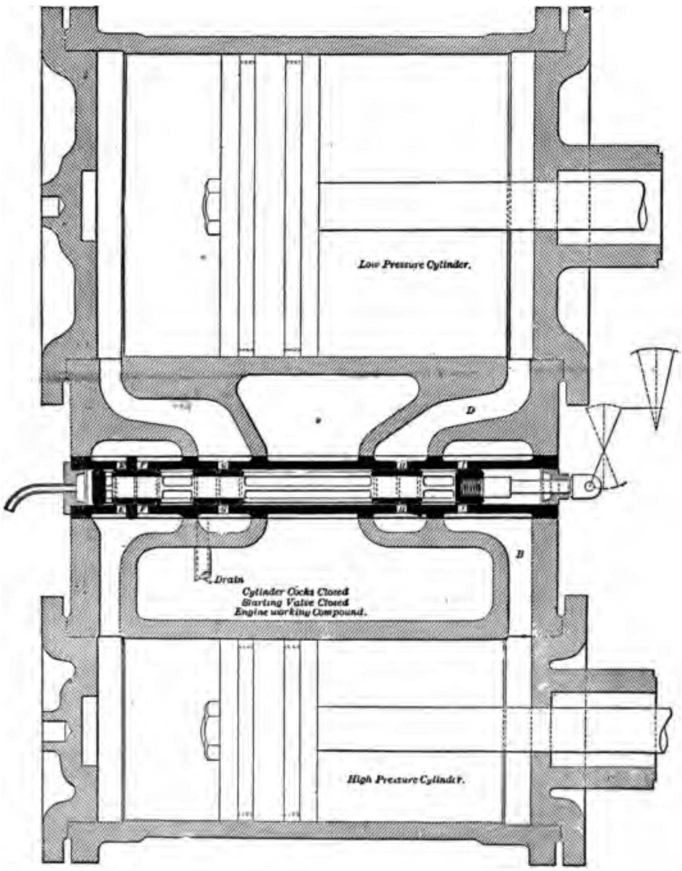


FIG. 122.
Starting Valve and Cylinder Cocks, Vauclain Type.
Starting Valve Closed and Cylinder Cocks Closed.

pistons is much greater when the engine is working with some h. p. steam entering the l. p. cylinder than when working compound.

The steam passing through the by-pass valve, when the engine is working "high pressure," acts just as a leak past the h. p. piston would act.

The operation of the starting, or by-pass valve, will be

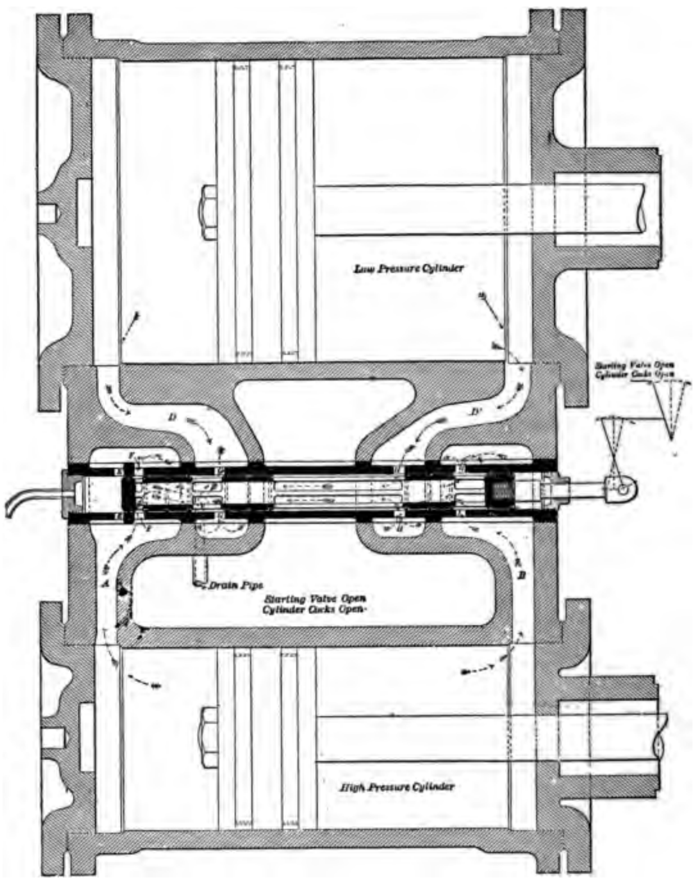


FIG. 123.
Starting Valve and Cylinder Cocks, Vauclain Type.
Starting Valve Open and Cylinder Cocks Open.

understood by referring to Fig. 121. On the right of the figure is a small diagram showing the positions of the lever in the cab, the full line corresponding to the position of the lever when the valve bears the same relations to the ports, as shown in the illustration. It will be seen that in this position of the valve, steam can pass freely from one end of the h. p. cylinder, through the ports, into the inside of the starting valve, and so on to the steam passage leading to the other end of the h. p. cylinder. This is the position of the valve when the engine is working with some of the h. p. steam passing to the l. p. cylinder.

Fig. 122 shows the position of the starting valve when the engine is working compound, all the ports being covered, no steam is passing through the valve. The full line in the diagram at the right shows the position of the lever in the cab, the right of the figure being toward the back end of the engine.

In Fig. 123 is shown the position of the valve and of the lever in the cab, when the cylinder cocks and starting valve are open. In this position there is free communication between both ends of both cylinders, and the cylinder cock drain pipe, through the centre of the valve. This allows the cylinders to be drained, as shown clearly by the arrows. But, of course, the drain pipe is lower than the h. p. cylinder, and not above it, as is here shown for the purpose of giving a readily understood explanation.

Figs. 124 to 126 show a new type of air valve and cylinder drain cock that has just been introduced for this type of engine. The experience with it is limited at this time, but it promises well, and is easily accessible. The body of the cock is in one casting, into which are put the two taper plugs, one of which, *X*, Fig. 125, controls the steam for starting, and the other, *Z*, controlling the l. p. cylinder cock. The passage leading to the cock *X* is connected to opposite ends of the h. p. cylinder, and those from plug *Z* lead to opposite ends of the l. p. cylinder. The two cocks have a squared end upon which is one arm which operates the two cocks simultaneously. In position No. 1. the plug X allows steam to pass through, putting in communication the opposite ends of h. p. cylinder, thence through valve to effective side of l. p. piston; all the openings in cock Z being closed. When the arm is moved to position No. 2,

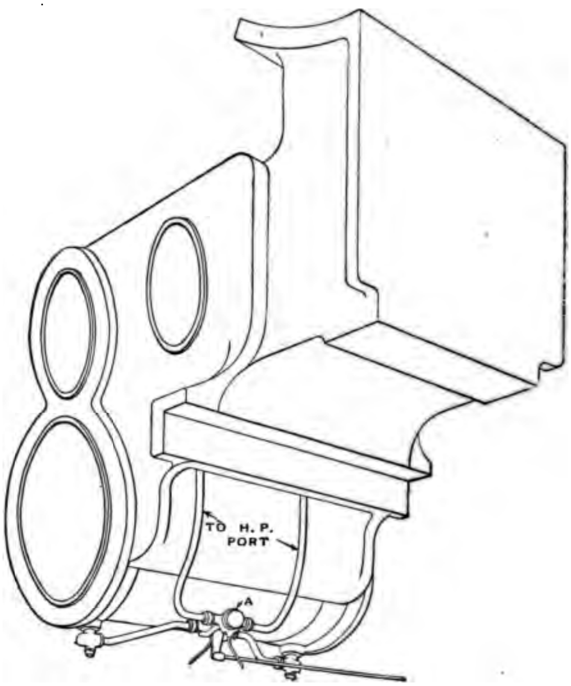


FIG. 124.
Recent Form of Starting Valve and Cylinder Cocks, Vauclain Type.

the opening in plug X allows the steam to pass through as before, but it also brings hole G opposite hole H, allowing any water to escape from h. p. cylinder to atmosphere. Plug Z, with arm in position No. 2, allows the three openings in the plug to come opposite the three openings in the body, thus draining the l. p. cylinder. The arm in position No. 3 closes all openings and is the running position. The

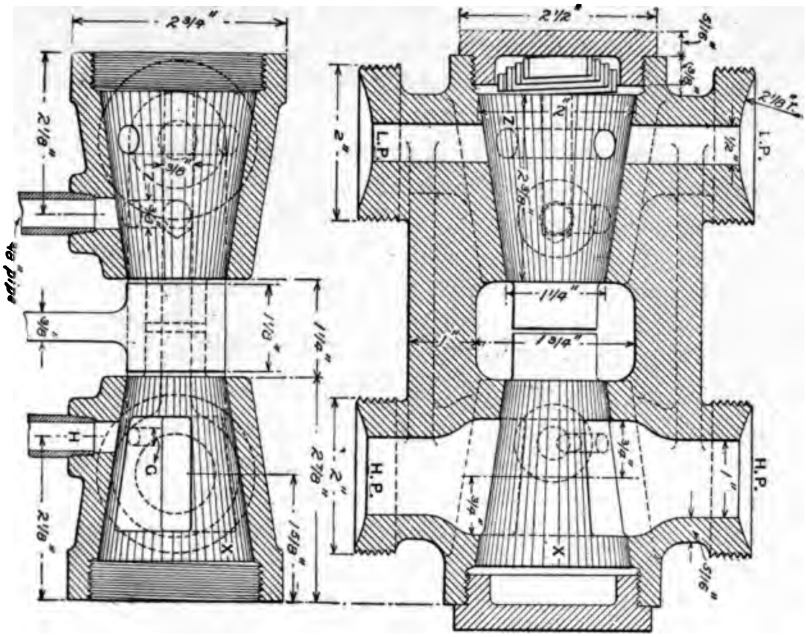
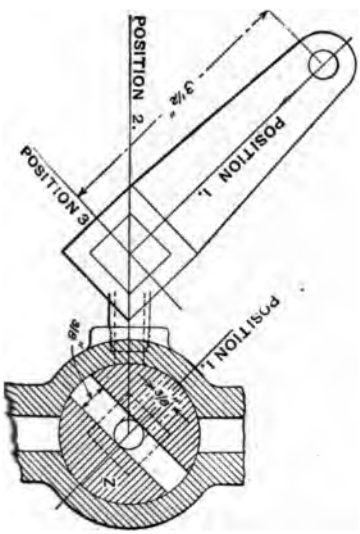


FIG. 125.



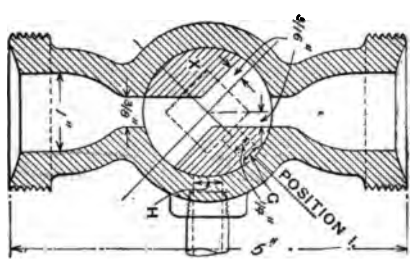


FIG. 126.

Details of Recent Form of Starting Valve and Cylinder Cocks, Vauclain Type.

cock is operated from the cab by a lever with a notched quadrant, corresponding to the three positions of the arm. The starting valve and cylinder cock is applied to the cylinder, as shown in Fig. 124.

121. Distribution of Pressure on Pistons.—The feature of this design, which at first glance would seem to be most open to criticism, is the connection to one crosshead of two pistons, of which the centres are about 18 inches apart and on which the total pressures must differ considerably. To determine the amount and variation of this difference of pressure with reasonable exactness an examination of a very large number of indicator cards taken simultaneously from both h. p. and l. p. cylinders would be necessary, and the inertia of the reciprocating parts must be taken into account. See Appendix P. Some knowledge of the subject can, however, be gained from an examination of the indicator diagrams shown in Figs. 11 and 12. The data for these diagrams is given in Table DD. The diagrams were divided into ordinates as shown in Figs. 127 and 128, and the difference between the forward pressure on one side of the piston and the back pressure on the other side was plotted for each ordinate, allowance being made for the piston rod areas. When the starting valve is opened these results will be materially altered. The inertia of the reciprocating parts was calculated for the different points of the stroke. See Appendix P.

From these results the curves shown by the heavy full lines in Figs. 127 and 128 were plotted. These curves indicate very closely the actual pressures on the crosshead, where the piston rods are attached for the h. p. and l. p. cylinders, at different parts of the stroke, the inertia of the reciprocating parts being taken into account. The numbers on the diagrams refer to the correspondingly numbered indicator cards of Figs. 11 and 12.

The full line on Figs. 127 and 128 shows the difference

TABLE D D.

Giving Data for Indicator Cards showing Steam Distribution on Baldwin Compound, on C. M. & St. P. Railway. See Figs. 127 and 128.

Card No.	Boiler Pressure, Gauge.	Revolutions per Minute.	Miles per Hour.	Cut-off in Inches.		Mean Effective Pressure.		Mean Effective Pressure, L. P. Cyl. Referred to H. P. Cyl.	Horse-Power.		Per Cent. of Work done by L. P. Cyl.
				H. P.	L. P.	H. P.	L. P.		H. P.	L. P.	
F 1	176	256	47.10	12.25	15.06	37.50	13.75	40.43	142.4	145.2	50.
B 2	176	256	47.10	12.25	15.00	41.25	12.50	34.75	141.4	127.3	47.
F 3	170	228	41.95	12.25	15.06	40.00	15.00	44.10	135.3	141.1	51.
B 4	170	244	44.90	13.25	15.94	51.88	15.00	41.70	169.4	145.6	46.
F 5	168	232	42.69	13.28	15.90	47.50	20.00	58.80	162.9	191.3	54.
F 6	174	140	25.76	13.28	15.90	64.50	25.00	73.50	134.0	144.5	52.
B 7	177	188	34.59	14.25	16.87	68.75	22.50	62.55	173.0	168.3	49.
F 8	175	192	35.33	15.41	17.62	70.00	28.75	84.52	199.7	227.7	53.
B 9	177	172	31.65	15.44	17.75	75.00	25.00	69.50	172.7	171.1	50.
B 10	171	156	28.70	15.44	17.75	78.75	27.50	76.45	164.4	170.7	51.
F 11	175	120	22.08	15.41	17.63	82.50	37.50	110.25	146.9	175.6	54.
F 12	170	80	14.72	15.41	17.63	81.25	38.75	113.93	96.4	127.8	57.
B 13	176	48	8.83	21.26	22.75	116.25	46.25	128.56	74.7	88.3	54.

between the actual pressure on the crosshead due to the h. p. piston, and that due to the l. p. piston. The vertical distances above the neutral line *A A* to the full curved line represent the excess of the total pressure on the h. p. piston above that on the l. p. piston. Distances below this line indicate how much the total pressure upon the l. p. piston exceeds that on the h. p. piston. The scale of pressures is given on the side of the diagram. It will be seen that the greatest difference in pressure is for the diagram taken at slow speed and late cut-off, and that for high speed and early cut-off the difference is comparatively small. Also that the effect of higher speed and lower initial pressure with the same cut-off is to greatly change the amount and distribution of the excess pressure. The tendency is to tip the crosshead, and hence to bring a bending load on the piston rods. It does not follow that this fact is an argument against the adoption of this design, but simply that a varying load, acting with a leverage of about 18 inches, and having from 300 to 600 reversals per minute at ordinary speeds, should be provided for in addition to the usual stresses on piston rods.

The lines on diagrams, Figs. 127 and 128 have the following signification: The dotted lines show the total

The indicator cards from which these diagrams were taken are from a ten-wheel Vauclain compound on the Chicago, Milwaukee & St. Paul Railroad, and the data regarding the cards is given in Table D D, the cards themselves are given by Figs. 11 and 12.

122. Advantages Claimed for the Baldwin Locomotive Works (Vauclain) System.—The claims made by the Baldwin Locomotive Works for the Vauclain compound, after an experience with about 400 engines, working under a great variety of conditions, are as follows.

These claims represent what this system of compound has been designed to accomplish :

1. To compound an ordinary locomotive with the fewest possible alterations necessary to obtain the greatest efficiency as a compound locomotive.

2. To develop the same amount of power on each side of the locomotive, and avoid the racking of the machinery resulting from uneven distribution of power.

3. To make a locomotive in every respect as efficient as a single expansion engine of similar weight and type.

4. To insure the least possible difference in the cost of repairs.

5. To attain the utmost simplicity and freedom from complication.

6. To realize the maximum economy of fuel and water.

7. To require the least possible departure from the methods of handling usual with single expansion locomotives.

8. To permit a train, in case of break-down, to be brought in without unusual delay, when using but one side of the locomotive.

9. To be equally applicable to passenger or freight engines.

10. To withstand the rough usage incidental to ordinary railroad service.

There are some who have had experience with this type of compound who would not certify to the justice of these claims, but the great majority of those who are using these engines believe that the Baldwin Locomotive Works have accomplished what they set out to do. It is noticeable that the claim is made that the engines are equally applicable for passenger and freight engines. As this is not true of any compound in existence, and cannot be, from the nature of things, it is not true for this type. So far as the mechanical construction is concerned, the statement is true; but in the matter of efficiency, it cannot be true, for no compound which has as much compression and wire-drawing as this, and other types at high speeds, can ever be relathe combined effect of the steam pressure and inertia of the piston and rod. The heavy full lines represent the difference between the pressures on the h. p. and l. p. pistons at

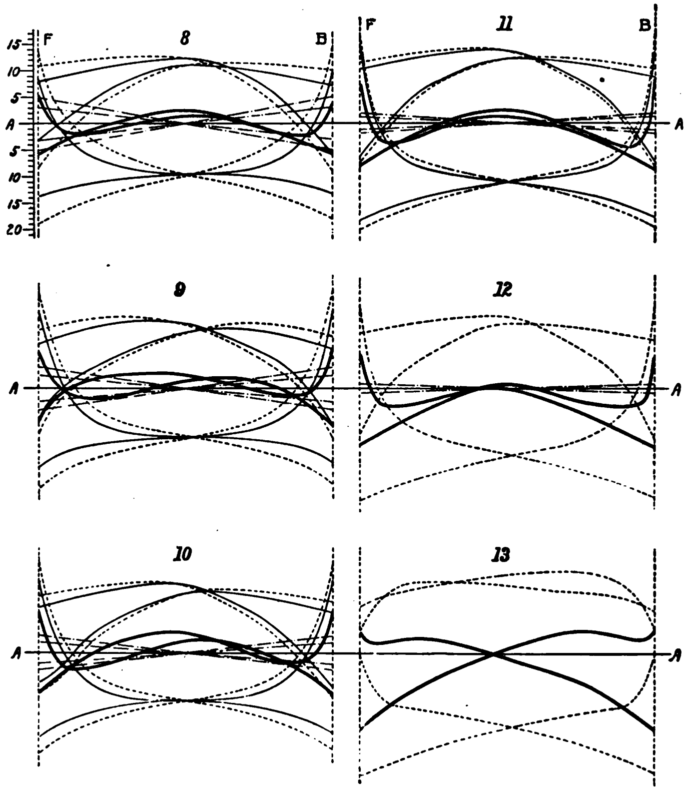


FIG. 128.

Diagrams showing Total Steam Pressure on High and Low-Pressure Pistons at Different Points in the Stroke of a Vauclain Compound. Also showing the Inertia of the Pistons and Piston Rods and the Total Comparative Pressures on the Top and Bottom of the Crosshead, Taking into Account the Inertia of the Pistons and Piston Rods.

The Scale at Side of Diagram is in Thousands of Pounds.

the crossheads. The scale at the side of the diagram indicates the total difference in pressure, and shows the amount of the twisting tendency on the crosshead for the different parts of the stroke and under different conditions.

The indicator cards from which these diagrams were taken are from a ten-wheel Vauclain compound on the Chicago, Milwaukee & St. Paul Railroad, and the data regarding the cards is given in Table D D, the cards themselves are given by Figs. 11 and 12.

122. Advantages Claimed for the Baldwin Locomotive Works (Vauclain) System.—The claims made by the Baldwin Locomotive Works for the Vauclain compound, after an experience with about 400 engines, working under a great variety of conditions, are as follows.

These claims represent what this system of compound has been designed to accomplish :

1. To compound an ordinary locomotive with the fewest possible alterations necessary to obtain the greatest efficiency as a compound locomotive.
2. To develop the same amount of power on each side of the locomotive, and avoid the racking of the machinery resulting from uneven distribution of power.
3. To make a locomotive in every respect as efficient as a single expansion engine of similar weight and type.
4. To insure the least possible difference in the cost of repairs.
5. To attain the utmost simplicity and freedom from complication.
6. To realize the maximum economy of fuel and water.
7. To require the least possible departure from the methods of handling usual with single expansion locomotives.
8. To permit a train, in case of break-down, to be brought in without unusual delay, when using but one side of the locomotive.
9. To be equally applicable to passenger or freight engines.
10. To withstand the rough usage incidental to ordinary railroad service.

There are some who have had experience with this type of compound who would not certify to the justice of these claims, but the great majority of those who are using these engines believe that the Baldwin Locomotive Works have accomplished what they set out to do. It is noticeable that the claim is made that the engines are equally applicable for passenger and freight engines. As this is not true of any compound in existence, and cannot be, from the nature of things, it is not true for this type. So far as the mechanical construction is concerned, the statement is true; but in the matter of efficiency, it cannot be true, for no compound which has as much compression and wire-drawing as this, and other types at high speeds, can ever be relatively so efficient, in comparison with single expansion engines, in passenger, as in freight service.

123. The Johnstone System on the Mexican Central Railway.—Figs. 129 and 130 show the construction of the Johnstone compound cylinder. The h. p. piston is in the centre of the annular l. p. piston which surrounds it. Between the pistons there is a double barrel made of cast iron. One of these barrels forms the h. p. cylinder; and the other, the outer one, forms the inner surface of the annular l. p. piston. The construction is clearly shown by the illustration. The ratio of the cylinders is generally 3 to 1. The valve, while

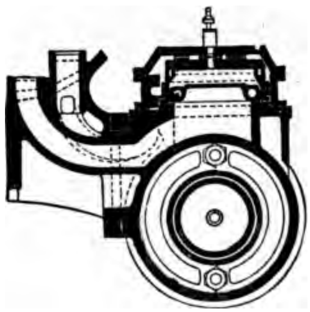


FIG. 129.
Cross-Section Through Johnstone Cylinders.

double, has but one valve stem. It is actuated by a link motion as usual. The outer section of the valve distributes steam to the h. p. cylinder, and the inner section to the l. p. cylinder. The inner section is loose within the outer section, and has a motion of about 1 inch independent of the outer section, for the purpose of giving a later cut-off in the l. p. cylinder, and to reduce the compression in the h. p. cylinder. The valve is cushioned from knocking by means of two springs, one on each side of the inner valve. The cut-off obtained by this arrangement is given in Table U 1. The starting valve used with this system is simply a three-way cock in the cab having a small pipe leading to the steam chest. Thirteen of these compounds are now in operation. The only change of any importance that has been made in the recent designs, is an increase of the steam

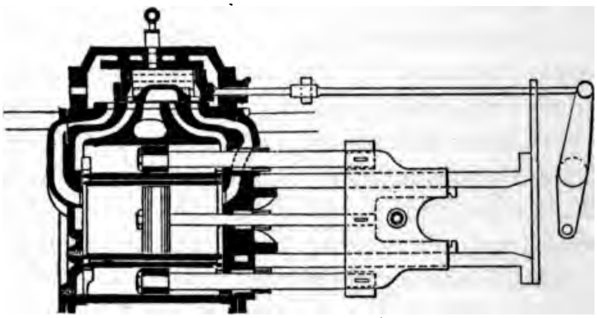


FIG. 130.
Arrangement of Johnstone Cylinders and Crosshead.

port which was done to reduce the wire-drawing and compression.

CHAPTER XIX.

DESCRIPTION OF FOUR-CYLINDER, TWO-CRANK RECEIVER COMPOUNDS—TANDEM RECEIVER TYPES.

124. Tandem Compounds on the Hungarian State Railway.—The Hungarian State railways have constructed some tandem four-cylinder receiver compounds with two cranks with the general construction shown in Figs. 131, 132 and 133. The general description of these engines is given in Table C C, Appendix R.

The service is that of hauling 160 tons at 50 miles an hour on a level. This is practically passenger work; in fact, this engine is used in passenger traffic. Like other receiver compounds with two cranks, the indicator cards have the same general appearance as those taken from two-cylinder receiver compounds. Such indicator cards as have been given to the public were taken at slow speeds. These give no indication of the amount of compression in the engines at high speed. Both h. p. cylinders exhaust in the same receiver. In starting, live steam is admitted to the l. p. cylinders and receiver by means of a starting valve which is opened by the reverse lever when in full forward gear. In this engine both valves are connected to the same valve stem, as in the Mallet system used on the Southwestern railways of Russia; 125. Likewise, as in the Mallet and Du Bousquet, 118, systems, the piston rod stuffing-boxes are packed separately and are accessible from the outside as distinguished from the Brooks tandem, 127. By connecting the two valves to one stem, it is necessary to raise the h. p. valve seat somewhat, as is clear from Fig. 131, but this does not give a larger clearance to the

h. p. cylinder than is common in compound locomotives. In order to reduce the weight of reciprocating parts as much as possible, the h. p. piston is forged on to the

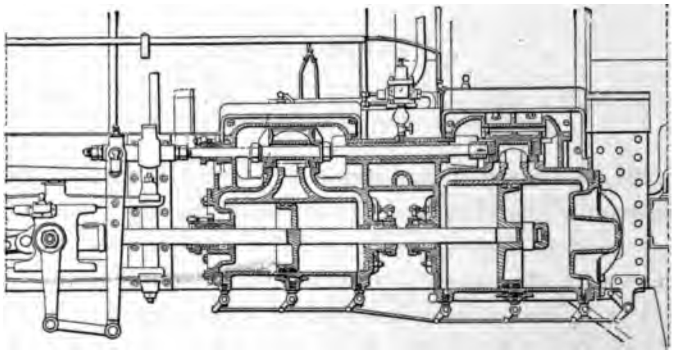
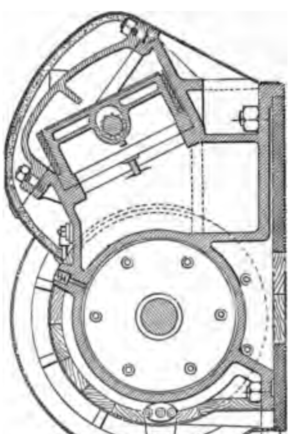


FIG. 131.
Tandem on Hungarian State Railways—Longitudinal Section.



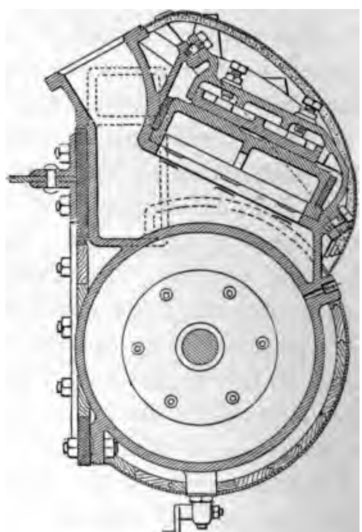


FIG. 132. FIG. 133.
Tandem on Hungarian State Railways—Cross Section.

piston rod, as shown in Fig. 131. The l. p. piston is keyed on, as shown. The valve chest and valve seats are inclined, as shown in Figs. 132 and 133. In the Mallet construction and the Brooks tandem, the l. p. cylinder is next to the crank, while in this engine and the tandems used on the Northern Railways of France, the h. p. cylinder is next to the crank. In the Hungarian type, the cylinders are cast in one piece.

125. Tandem Compounds on the Southwestern Railways of Russia.—In this system the l. p. cylinder is placed next to the crank. Both valves are connected to the same valve stem, but are independently connected, as shown in Fig. 134. The l. p. piston is attached to the piston

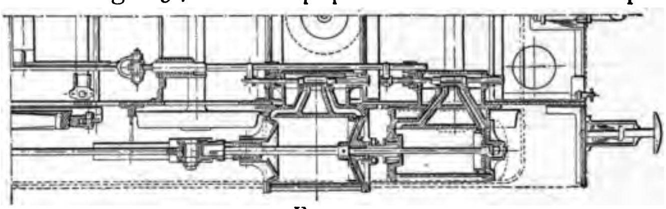


FIG. 134.
Mallet Tandem on Southwestern Railways of Russia—Longitudinal Section.

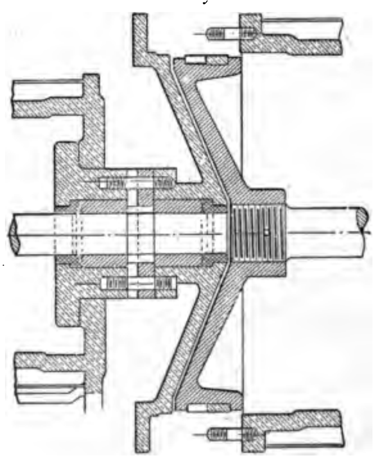


FIG. 135.
Piston for Mallet Tandem.

rod by screw-thread and pin, as shown in Fig. 135. The stuffing-boxes are accessible from the outside. The clearance in this design is considerable, as the valve seats are unusually high. The reciprocating parts have been made light in weight by the use of single plate pistons. The Mallet starting gear is used, but not of the type that is used for two-cylinder engines. This system has a simple valve which admits steam to the l. p. cylinder and receiver at starting whenever the reverse lever is in full forward gear. The construction of this engine in the cylinder part is clearly shown in Figs. 134 and 135. The general dimensions are given in Table C C, Appendix R.

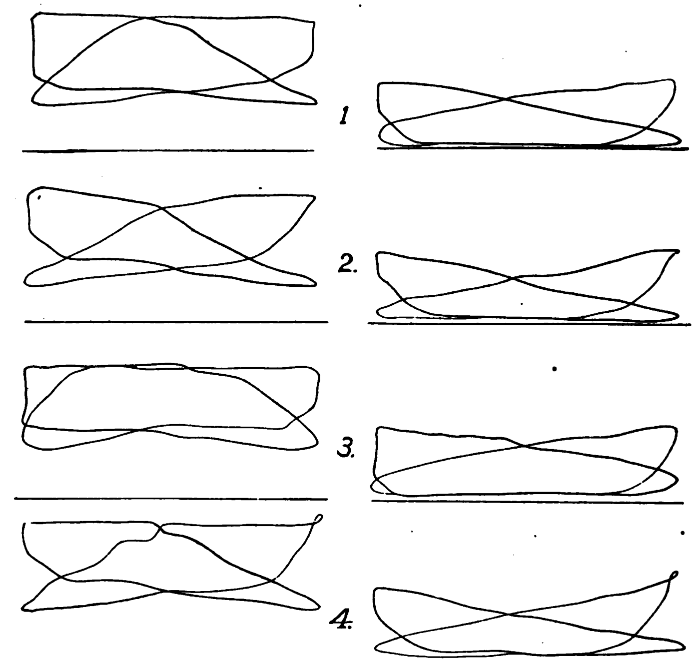


FIG. 136.
Indicator Cards from Mallet Tandem on Southwestern Railways of Russia.

126. Indicator Cards from Tandem Compounds on the Southwestern Railways of Russia.—Some indicator cards taken at a comparatively slow speed are given in Fig. 136, and the data about the card will be found in Table E E.

TABLE E E.

Giving data for Figs. 134, 135 and 136 for Mallet Compound on South-western Railways of Russia. Cylinders 13 in. and 19.7 × 23.6 in. Drivers 79 in. diameter.

Card Number.	Boiler Pressure. Pounds.	Revolutions per Minute.
1	162	79
2	161.3	120
3	165.	101
4	165.	168

127. **The Brooks Tandem System.**—Figs. 137 to 144 show the details of the tandem four-cylinder compound recently brought out by the Brooks Locomotive Works of Dunkirk, New York. The general dimensions of the engine are given in Table C C. Fig. 137 shows the end view and half section through the l. p. cylinder, also shows the reducing valve which admits steam from the steam pipe directly to the l. p. cylinder at starting. Fig. 138 shows a section through the cylinders longitudinally. The h. p. cylinder is placed ahead of the l. p. The l. p. valve is of the ordinary pattern plain slide valve. The h. p. valve is of the piston type. These valves have an opposite motion; that is, when the h. p. valve goes ahead the l. p. valve goes back. This reversed motion is produced by means of the rocker arm shown in Figs. 138 to 140. The h. p. steam pipe has a 2 inch vacuum valve and the l. p. cylinder has a 2 inch relief valve. The h. p. valve seat is made in the form of a bushing shown in Figs. 141 and 142. The rock arm bearing for reversing the motion of the valve is oiled by means of a hole drilled through the centre of the bearing, as shown in Fig. 140.

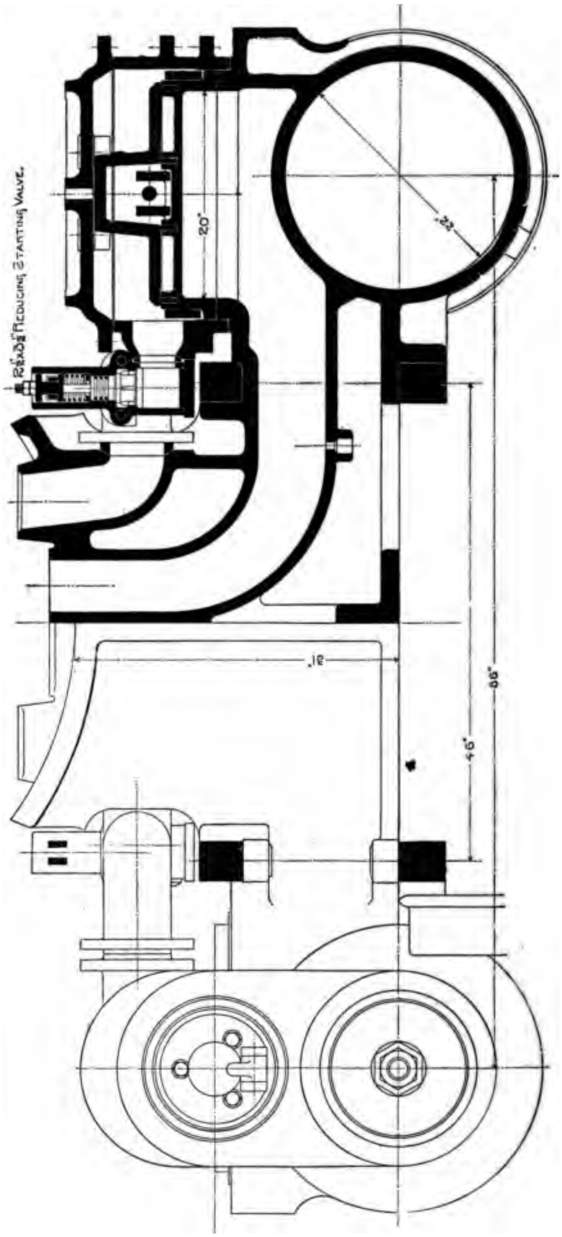


FIG. 137.
Brooks Tandem—Cross Section Showing Cylinders and Steam Chests.

The exhaust steam from the front end of the h. p. cylinder reaches the receiver through the centre of the h. p.

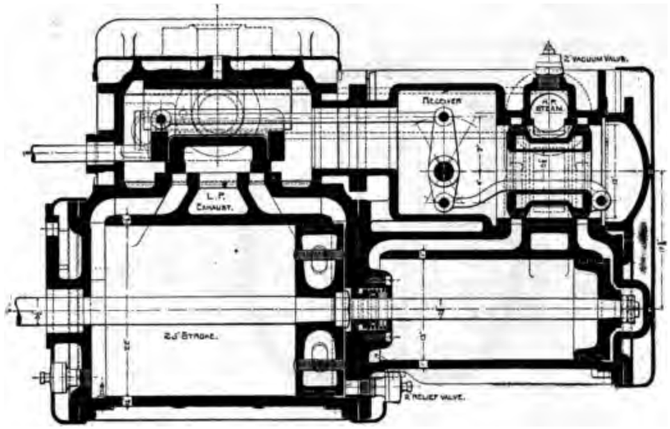


FIG. 138.
Brooks Tandem—Longitudinal Section.

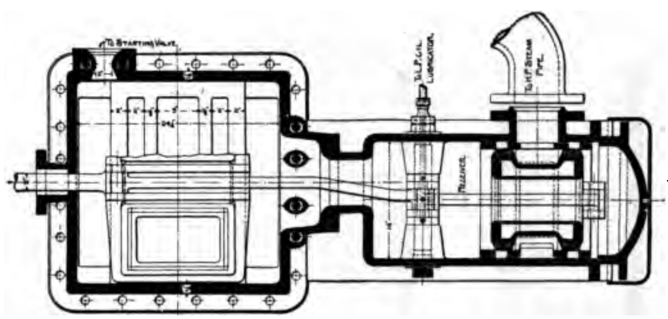


FIG. 139.
Brooks Tandem—Plan of Steam Chest.

valve, as is evident from Figs. 138 and 139. This hollow h. p. valve has a lining of wrought iron pipe, as shown; the space between the pipe and the valve being filled with asbestos. This valve has removable ends to facilitate the insertion of packing rings.

Figs. 143 and 144 show the starting valve arrangement. This valve has a spring which keeps it closed under normal conditions, as shown in Fig. 143, but whenever the

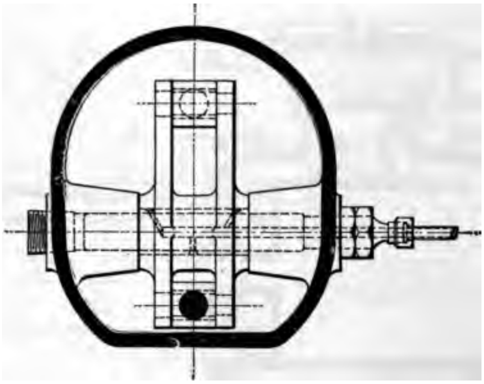


FIG. 140.
Brooks Tandem, Valve Rod Rocker.

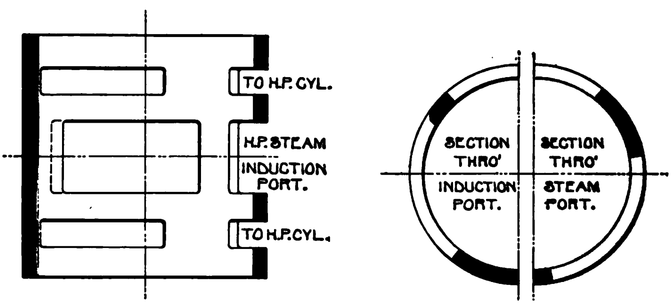


FIG. 141. FIG. 142.
Brooks Tandem. High-Pressure Valve Bushing.

valve motion is thrown into full forward or backward gear, the projections on the rod connected to the reverse shaft arm force the spring down and the valve open, as shown in Fig. 144. In this way steam is admitted to the l. p. cylinder direct only when the reverse lever is in full forward

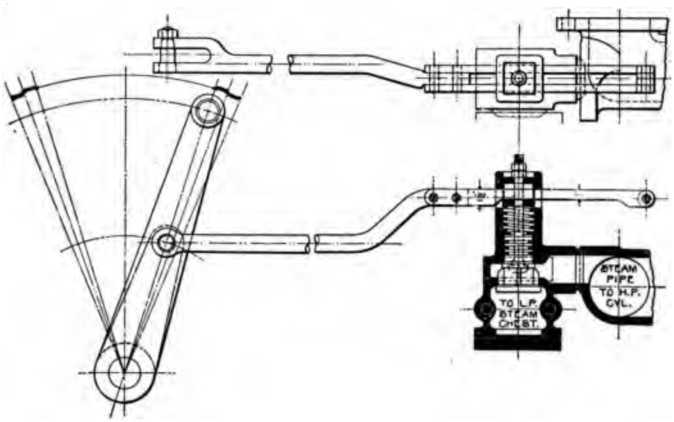


FIG. 143.
Brooks Tandem—Starting Valve Connections.

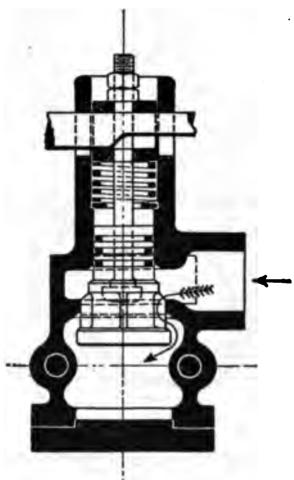


FIG. 144.
Brooks Tandem—Starting Valve.

or back gear. The connections for the valve and the combined stuffing-box are shown in Figs. 138 and 139.

CHAPTER XX.

DESCRIPTION OF THREE AND FOUR-CRANK COMPOUNDS.

A discussion of the elementary features of these types of compounds will be found in Appendixes I and K.

128. **Webb System; Express Locomotives without Parallel Rods.**—The general arrangement of the cylinders and steam connections of compound locomotives of the Webb system is illustrated by Fig. 145. In this figure *h h*

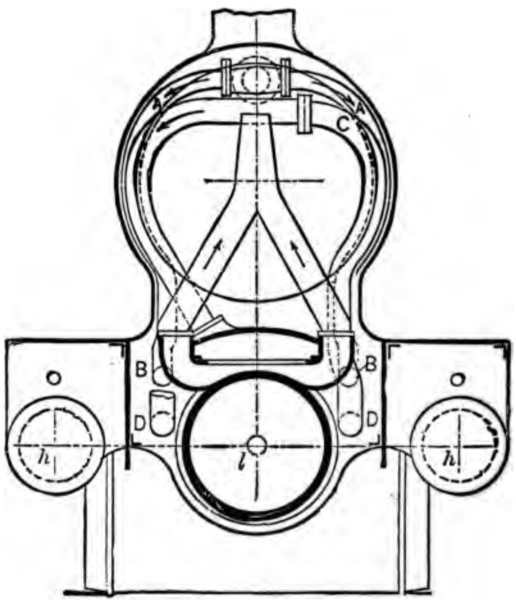


FIG. 145.
Webb Three-Cylinder Compound—Cross Section Through Cylinders.

are the h. p. cylinders, which are placed so that the centres are in a transverse line about four feet back of the front tube sheet, and which are connected to the second pair of driving wheels. The l. p. cylinder *l* is placed beneath the smoke box, and is connected to the forward pair of driving wheels. The course of the steam from the boiler is through the pipes *A A* to *B B*, and thence back to the h. p. cylinders. The exhaust from these cylinders is led through the pipes *D D*, and thence around the smoke box through the two pipes *C* to the l. p. steam chest. The course of the exhaust from the l. p. cylinder is clearly indicated in the figure. The disposition of the cylinders and steam pipes is essentially the same in the Webb compounds for passenger and freight service. The most noticeable peculiarity of the system is the absence of driving connection between the h. p. and l. p. axles, there being no coupling rods on engines having two pairs of driving wheels.

129. **Webb System; Freight Locomotives with Parallel Rods.**—In one design for freight service there are three driving axles, the first being driven by the l. p. cylinder, and the second and third, which are coupled, being driven by the h. p. cylinders. It will be seen that even in this case there is no connection by coupling rods between the h. p. and l. p. cylinders. The principal dimensions of a recent Webb compound locomotive are as given in Table C C, Appendix R.

130. **Webb System on Pennsylvania Railroad.**—A compound locomotive of this type was purchased by the Pennsylvania Railroad and put into service in 1889. The results of practical trial with the heavy trains used in the United States were satisfactory in economy, but unsatisfactory in hauling power. It has been found difficult to start the ordinary weight of train with this engine, owing to the slipping of the drivers, which were not provided with parallel rods. When the trains are light the engine works with the most excellent economy, and shows a decided saving in fuel. The reports from the London & North-

western Railroad of England, where these engines have been principally used, are very complimentary.

131. Three-Cylinder System Used on the Northern Railways of France.—A compound locomotive having one h. p. cylinder and two l. p. cylinders was built by the Northern Railway of France, and exhibited at the Paris Exposition in 1889. The general arrangement of the cylinders and steam connections of this locomotive is shown in Fig. 146. Referring to this figure, *h* is the h. p. cylinder; *l l* are the l. p. cylinders; *A* is main steam pipe to the h. p.

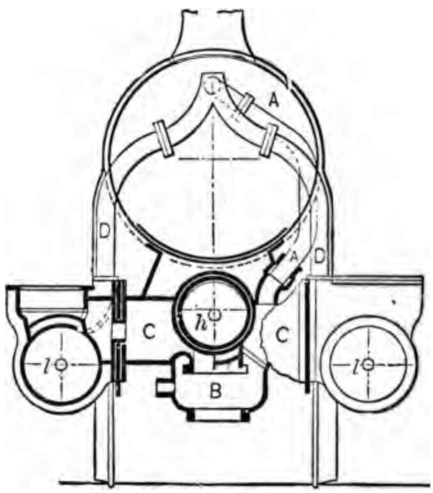


FIG. 146.
Three-Cylinder Compound on Northern Railways of France.

cylinder; *C C* is the receiver; and *D D* are the l. p. exhaust pipes. The l. p. cylinders are placed as usual, and have the valve chests above. The h. p. cylinder is placed below the smoke box with its valve chest *B* below it, and is inclined to an angle of one in ten. The locomotive is of the Mogul type, having six coupled driving wheels, the middle axle being the main driving axle for all three cylinders. The l. p. cranks are at right angles, and the h. p.

crank is midway between them, thus making an angle of 135 degrees with each l. p. crank. It will be noticed that the receiver is formed in the cylinder castings, and not by pipes, as in the locomotives previously illustrated.

132. **Valve Gear for Three-Cylinder Compound on Northern Railways of France.**—The h. p. valves are a special feature of this engine. These consist of a main valve and a cut-off valve, which slides on the back of, or below, the main valve, the whole forming a combination which in principle is the same as the Meyer and Ryder cut-offs. The edges of the cut-off valve form an oblique angle with the axis of the cylinder, as in the Ryder valve gear, and the ports in the main valve are correspondingly inclined at the back of that valve, but are twisted so that on the face next to the cylinder they are placed as is customary. The edges of the exhaust port in the cylinder casting are, however, inclined, and the exhaust cavity in the main valve is formed to correspond. The yoke which drives the main valve does not fit it at the sides, and so permits a transverse movement while controlling it longitudinally. A second yoke incloses the valve, and permits a longitudinal movement, but holds it transversely. This yoke is connected to a stem, which passes through a stuffing-box in the side of the valve chest, and is operated from the cab by lever connections. It is clear that the h. p. cut-off can be adjusted at any time by means of this connection, while the valve is so proportioned that in its extreme position the steam and exhaust ports remain open for all positions of the h. p. piston, and steam is thus allowed to blow through the h. p. cylinder without doing work. The engine can, therefore, be started by the l. p. cylinders with steam from the boiler, the h. p. piston being then practically inoperative; and as the l. p. cranks are at right angles, the starting conditions will be the same as for a single expansion locomotive.

133. **Summary of Three and Four-Crank Compounds.**—It has been shown by a large number of examples that the four-cylinder two-crank types can be made perfectly practicable in regular service and with outside connections, and for this and other reasons it is evident that three or four-cylinder compounds with more than two cranks will never be generally used, and, therefore, a consideration of the theoretical economies of such engines has been omitted here. In Appendixes I and K will be found a discussion of some features of such three-cylinder three-crank and four-cylinder four-crank compounds as have been built up to this time.

Such other three and four-cylinder compounds with three or four cranks as have been designed have not been raised to sufficient prominence to make it desirable to discuss them here.

134. **Miscellaneous Designs of Compounds that have not been Put in Service.**—Besides the designs already shown, a great many have been proposed, such as the Strong, Wright, Ball, Weir, and others; but as these engines are more or less complicated (some of them are exceedingly complex), and have never been put into practical operation, their consideration is omitted here for lack of space, and also because the practical value, whether good or bad, of such designs has not been demonstrated by actual construction.

CHAPTER XXI.

SUMMARY ABOUT STARTING GEARS.

135. **Automatic Starting Gears with Intercepting Valves.**—These gears are used solely with two-cylinder receiver compounds, and have been described in Chapter XV. The starting power with these gears is at a maximum at some point during the first revolution. This maximum point may even occur during the first quarter, and thereafter the starting power decreases until it becomes the same as when the engine is compounded. That is, the receiver becomes so charged with exhaust steam from the h. p. cylinder that the automatic intercepting valve opens and the engine works compound almost immediately after starting. This change may take place any time after the first exhaust from the h. p. cylinder. If the receiver is large, say four times the volume of the h. p. cylinder, the change to compound may not take place until two or possibly three exhausts have been made, but, ordinarily, it will take place about at the end of the first half revolution. After this the engine works compound, and with a greatly reduced hauling power. It is clear that the larger the receiver the greater will be the number of exhausts required to fill it, so that, with large receivers, the period of increased starting power will be prolonged. Such engines as have been built with these gears have worked well in practice when they were well designed, except in such cases as have required a long continued heavy pull at starting. Under these conditions, this type of starting gear has proved inadequate. For

pulling long trains out of a siding, or hauling heavy loads up a hill, or starting on a hill, or for starting heavy close-coupled vestibule trains, this type of gear is unsatisfactory; but for all average work it has been shown to be quite practicable. As might be supposed, from the development of other features of locomotives, it is the unusual condition, not the average, which controls, and, therefore, in starting gears for compounds it has been found necessary to take into account the maximum and unusual requirements rather than the average. Perhaps it is for this reason that recent designs of starting apparatus of this class—automatic intercepting valves—have been given an "emergency" feature which permits the engineer to run with a separate exhaust for the h. p. cylinder when it is found desirable to do so, to save time or to haul a few additional cars over a bad place. The Richmond Locomotive Works (Mellin) gear is an example of this kind.

From recent developments it is quite clear that there is now a tendency, even with those who formerly favored automatic gears, to give the engineer such apparatus as will enable him to run non-compound when starting at slow speeds for a sufficient time to enable him to get control of the train. Mr. von Borries, who has been a strong advocate of automatic starting gear of the kind that permits only one revolution at the most before automatically changing to compound action, has quite recently decided to use a new arrangement, on all future compounds, which will permit the engine to be run non-compound a sufficient length of time to enable the engineer to get control of the train. This is especially important by reason of the wide experience of Mr. von Borries with two-cylinder receiver compound locomotives, and from the fact that he was the original inventor of the automatic intercepting valve.

From what is now before us, it appears that Mr. Mallet's original plan of placing the compound and non-compound action at the will of the engineer is coming to the front for future use. At the first introduction of compounds, railroad men feared to give engineers control of the compound action, and therefore favored automatic intercepting valves; and also it was not thought advisable to give the engineer any more handles to turn or duties to perform than he already had; but now that all are more familiar with compounds, and the advantages of compounding are better appreciated, there is a general tendency to make engines satisfactory at starting and at all other times, even with the probability of requiring engineers to exercise better judgment and to do more manual labor when starting out with a heavy load. This seems a very logical conclusion, and will probably lead to simpler designs hereafter, and further, this will give a two-cylinder receiver compound that will start trains with quite as good satisfaction as the four-cylinder non-receiver type, and generally better than the single expansion locomotive. Having this in mind, it is pretty clear that the future will see less automatic and more non-automatic starting gears for two-cylinder receiver compounds.

136. **Automatic Starting Gears Without Intercepting Valves.**—These gears, of which the Lindner is the best known example, give practically the same maximum power at starting as the automatic intercepting valve type, but have the advantage of being very much simpler. In fact, they are the simplest of all types. All that has been said in 135 about automatic gears applies with equal force to this type. They work well under average conditions, but the unusual demands for hauling power are not adequately provided for.

137. **Non-Automatic Gears With Intercepting Valves and With Separate Exhausts for the High-Pressure Cylinders.**—This type of starting gear is not adapted to permit a separate exhaust for the h. p. cylinder

for any considerable speed, but is intended solely to provide starting power for unusually heavy demands. If used continuously, there is a decided loss of efficiency of the engine, and, in some designs, the fire is badly torn up by the force of the blast. It is intended to be used with discretion, and will give greater hauling power to the compound at slow speed than is possessed by a single expansion engine of equal rating. This greater power is given by the larger dimensions generally used for compounds for both h. p. and l. p. cylinders.

138. **Starting Gears for Four-Cylinder Compounds.** —When four cylinders are used, whether the engine be a four or a two-crank locomotive, or with or without receiver, there is practically a duplication of the cylinder power on each side, and if steam be admitted from the boiler to the l. p. cylinder and receiver and at the same time to the h. p. steam chest, the locomotive will have greater starting power than a single expansion engine of equal rating. One l. p. cylinder will be always ready to act with great power and generally one h. p. cylinder will be in such a position as to assist. Starting gears for this class of engine need no especial attention, and intercepting valves are unnecessary. Generally a small valve is provided for admitting steam into the receiver from the steam pipe whenever there is steam therein after the throttle has been opened, but only when the reverse lever is in full gear. When the reverse lever is hooked back one notch, the valve is closed automatically. In this way the engine can be run with increased power by admitting steam directly into the l. p. cylinder as long as the reverse lever is allowed to remain in full gear. The steam supply at this time to the l. p. cylinder direct from the steam pipes is but through a small pipe, and as the speed increases the wire-drawing through this pipe increases and a much smaller amount of steam is used in this way per stroke after the train is moving. But it is necessary to close the

direct supply valve and not permit it to be used at all times, otherwise the economy will be seriously affected. This has led to a demand for automatic closure when the reverse lever is hooked up one notch.

CHAPTER XXII.

REASONS FOR ECONOMY IN COMPOUND LOCOMOTIVES.

139. **Possibilities of Savings.**—Compound locomotives are considered to be more economical than single expansion locomotives for all of the common reasons why compound engines generally are more economical than single expansion engines, and for some additional reasons.

The principal claims for better efficiency are based upon:

(a) Greater expansion of steam, 45–52.

(b) Less condensation of steam due to lower range of temperature in cylinders, 69–72.

(c) Incidental saving due to better action of the blast on the fire, and the somewhat decreased rate of combustion in the firebox, 83, 142–145.

Other reasons than these are frequently given; but the possible saving outside these features is too small to be considered at this time when locomotives are designed with so little attention to loss of heat and are operated with such reckless disregard of efficiency, 70, 145, 147.

The possibilities of saving by compounding are pretty clearly shown by indicator cards. It can be determined within 5 per cent., from an examination of indicator cards taken from a single expansion locomotive, what would probably be the saving from compounding, provided the design of compound is assumed to be the best that can be devised with the present knowledge. Many of the reported savings from compounding have resulted from unfair comparisons in which no allowance was made for the advantages given to the compound, such as higher steam pressure, larger grate area, and increased heating surface. As this is now well understood by railroad men, reported savings at the present time are looked upon with suspicion, and this it is, perhaps, as much as anything else, which recently led the American Railway Master Mechanics Association to appropriate a considerable sum of money to carry on a laboratory investigation of the relative merits of compound and single expansion locomotives at the Purdue University, on the plan already commenced by Professor Goss of that university. The theories of economy due to compounding are so complex and involved in their nature that mathematical investigation is practically valueless until certain factors are definitely determined, and it is useless to theorize much in detail about the value of compounding locomotives until more accurate data is at hand.

140. **Saving by Greater Expansion.**—In order to gain greater expansion, the wire-drawing common in locomotives must be greatly reduced. This demands better valve motions and larger ports and passages, and if the possibilities of gain in expansion are to be fully utilized, the steam pressure must be considerably increased. With our present knowledge, and our method of oiling cylinders, 200 pounds per square inch above the atmosphere is almost the limit of boiler pressure for good practical service, 7–19.

The saving due to compounding must very largely result from a gain in expansion, and the more perfect use of the higher potential of increased steam pressures. When compared with a single expansion locomotive, the saving of the compound will vary almost directly with the gain in the useful work from a given weight of steam by greater expansion. This especially applies to the substitution of compounds for single expansion locomotives for use in slow freight service on heavy grades and for suburban passenger work. For high speed passenger work, the greater loss, almost universal so far, in compounds, from wire-drawing and compression, greatly reduces the saving otherwise possible.

It may not be clear at first why the same expansion cannot be obtained in single expansion locomotives as in compounds, but on reflection it will be seen that; the mechanical difficulties with the valve motion at short cut-offs; the very uneven turning power applied to the drivers, when large cylinders are used, and the enormous cylinder condensation at short cut-offs, compel the use of compounds for high grades of expansion. With cylinders large enough to furnish the needed expansion, all in one cylinder, the tendency to slipping drivers is so great as to prohibit the use of much expansion in a single expansion locomotive.

141. Saving by Reduction of Condensation.—To gain the saving resulting from less condensation due to lower range of temperature in the cylinders, better insulation of the steam passages, steam chests and cylinders must be had, and no great gain by saving in condensation may be expected unless good heat insulation is provided, 69–72, 151–152.

142. Saving by more Complete Combustion.—To gain the incidental saving due to a better action of blast, there must be a proper arrangement of smoke box apparatus. At the present time no one seems to know, because of lack of accurate data, just how to arrange the smoke box mechanism to get the best results. The practice on different roads varies altogether too much to indicate any uniformity in opinion. All know how to get a fairly good draft with a given exhaust, and this can be done in several ways. At the present time each new lot of locomotives is experimented with until a sufficient blast is obtained, and there the matter is dropped. Hence, at the present time no safe directions can be given for the location of smoke box apparatus, and the designer will have to be guided by the prevailing practice on the road for which the designs are made and make such changes after the engine is completed as will give satisfactory results. This much neglected matter is now being investigated by the American Railway Master Mechanics Association, 145–147. See Fig. 147.

143. Saving in Fast Express and Passenger Service.—It is only under the best conditions that much saving can be expected from compounds in fast passenger work. That a practical saving is possible in this service must be admitted by all who have studied closely the large theoretical saving possible with compounds. Without doubt a decided saving will be found in fast service when the valve motion receives more attention and the steam pressure is raised to about 200 pounds above the atmosphere. In some instances a saving in fast service has already been reported, and it is undoubtedly true that the reported savings were found, but whether the economy resulted from the inferior action of the single expansion engine with which the compound was compared, or from the fact that the passenger service was so slow and heavy as to give the compound somewhat the same advantage that it has in freight service, is not known. Really accurate tests of compounds in passenger service have never been made, and ordinarily accurate tests have only been made in one or two instances. See Appendixes M and N. An average of the more reliable results obtained shows no decided advantage for the compound in fast service, but this may result from the inferior action of the steam regulating apparatus, 12–19. There is no proof, however, that would lead to a safe conclusion that compound locomotives of the best design now built, or when built with the best obtainable knowledge, are not more economical than single expansion locomotives in passenger service.

144. Saving in Slow Grade Work and in Freight and Suburban Service.—The possible saving in slow, heavy freight service with equally good designs and equal advantages in all other respects for compound and single expansion locomotives, varies from 15 to 50 per cent. according to the conditions of operation. In general, the harder the engines are worked, the greater will be the saving from compounding, as by it a more complete utilization of the power of the steam will be obtained by greater expansion, 5, 12–19, 145.

There are special cases where incidental economies that have not been mentioned here may be expected; one of these is in elevated and suburban service where it is necessary to use mufflers on the exhaust of single expansion engines. These mufflers produce a back pressure varying from 10 to 20 pounds per square inch on the pistons depending upon the condition of the mufflers. They quickly become clogged with carbonized cylinder oil and cinder from the smoke box and the perforations are reduced in size so much as to require boring out frequently. With the compound a wide open exhaust nozzle is used, as the final pressure of the exhaust is reduced, and there is less noise than with a single expansion engine equipped with mufflers. The saving in fuel by reduction in back pressure probably amounts to as much as the saving from compounding itself. Such incidental savings as this and also some incidental losses, depend upon the conditions, and an estimate of a probable saving by the use of compounds can be made only when all of the conditions are known, and therefore each case should be studied by itself, 139–147.

145. **How Saving is Affected by the Price of Fuel and Rate of Combustion.**—The percentage of total train expenses that will be saved by compounding depends largely upon the cost of fuel. If the compound is well adapted for the work it has to do there will be some advantages incidental to the less amount of fuel burned in a given time; it will be easier for the firemen, and there will be some reduction of repairs to the engine, but the main portion of any money saving must be expected

from the actual saving in the fuel account. Where coal is cheap, say from 90 cents to $1.50 per ton, the saving per year in dollars and cents due to compounding will not be very great, but where coal costs from 7 to 10 dollars per ton,

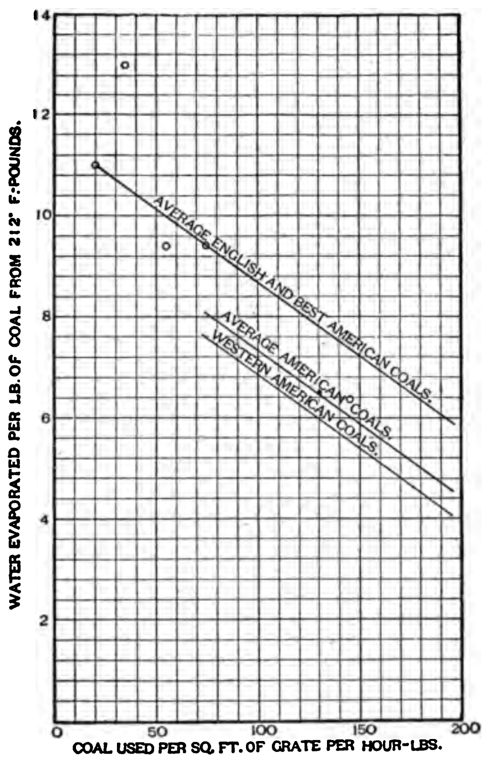


FIG. 147.
Diagram Showing the Relative Values of Different Fuels and the Increased Boiler Efficiency with Low Rates of Combustion.

as in some of the Western States, or from 12 to 22 dollars per ton as in Mexico and the southwestern United States, the money saving due to compounding is very great, and amounts to more than the maintenance, deterioration and interest cost for the entire locomotive equipment. This has been the experience on the Mexican Central Railroad, where the price of coal varies from 18 to 23 dollars per ton. This has also been found to be true in Austria, where coal is expensive and of inferior quality, that will give an evaporation of only 3½ pounds of water per pound of coal.

Fig. 147 shows the relative value of different kinds of coals when burned with different degrees of draft. That is, it shows how the efficiency in water evaporation in a locomotive boiler decreases as the coal used per square foot of grate per hour increases. Incidentally it also shows the difference between English and American coals and the greater value of good fuels.

Fig. 148 shows the decrease in relative cost of fuel and other train expenses per ton mile of cars and lading with different prices for fuel, as the trains are increased, and will be found useful in reaching a conclusion about the value of a compound locomotive on any given road. A railroad manager knows that where fuel is cheap even a large saving in weight of fuel will but little effect the total cost of train expenses. The ratio of the fuel expenses to total train expenses is given by Fig. 148. It is evident from this diagram that outside of any advantages that may accompany the use of compounds, in the way of reducing repairs and decreasing the labor of the firemen, there is but a small percentage of total train expenses to be saved by compounding when coal is about one dollar per ton; but, on the other hand, it is also clear that where coal costs from 15 to 20 dollars per ton, a 20 per cent. saving by compounding very materially reduces the total train expenses.

The need of compounding is greater on some roads than on others. Where the locomotive equipment is old-fashioned, small and overworked, and where the boilers have small grates, the introduction of modern heavy compound locomotives with large grates frequently brings a saving in total train expenses amounting to 40 per cent. This arises from the fact that heavier trains are hauled with the same train crew and much less fuel per ton mile is used ; but such savings are not due to compounding alone but to

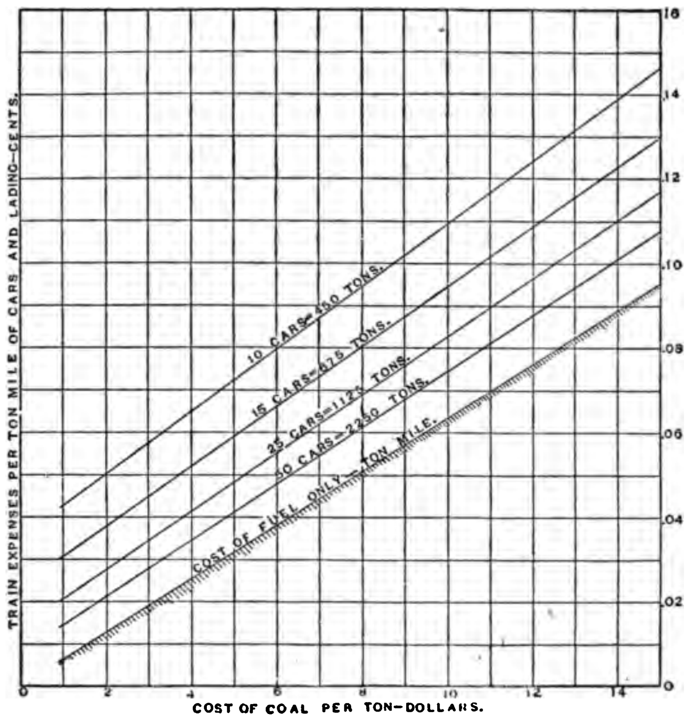


FIG. 148.
Diagram Showing the Comparative Cost of Fuel and Other Train Expenses for Varying Prices of Fuel.

the combined effects of heavier engines, larger grates and heating surfaces, and the saving due to compounding. Compounding generally gives also some indirect advantages which should not be overlooked, for instance it is not possible to give to large locomotives the same relative boiler capacity that is given to small locomotives, and the larger and heavier the locomotive the smaller is the relative boiler capacity that can be provided. There is a limit in the increase of grate area at which a fireman cannot fire properly, and beyond that a further increase gives no advantage. It is at these limits of increase of grate area and steam-making capacity that the advantage of compounding by reduction of total amount of fuel used in a given time is of great benefit. At the present rate of increase of total train loads it is quite clear that the time will soon be at hand when compounding of locomotives will have to be resorted to in order to reduce the demand on the boilers. See Fig. 147.

Mr. Axel S. Vogt, Mechanical Engineer of the Pennsylvania Railroad, in summing up, recently, the probable advantages of compound locomotives for future work, has said in effect that if the weight and speed of trains continues to increase, a limit of grate area will soon be reached, beyond which the fire cannot be properly managed, and one possible result of this will be to require better use of the steam which is made. And any further increase of weight and speed of trains will necessitate the introduction of more improved methods of utilizing the steam, so that it appears that the compound system will eventually be used to reduce the demand on the boilers.

146. **Cost of Repairs.**—To offset the saving by the use of compound locomotives, there is some extra cost of maintenance. The additional first cost will range from 100 to 500 dollars per engine, depending somewhat upon the size, but mostly upon the design. The actual cost of the additional parts for compounding will probably not be over 200 dollars per engine, for either the two-cylinder receiver type or the four-cylinder non-receiver type ; when but two cranks are used. The additional cost of three and four-cylinder types with receivers and with three or four cranks will be considerably greater. If the steam pressure on the compound is higher than the single expansion engine with

which it is compared in cost, the compound will cost something more for the stronger boiler that will be necessary, but this addition is not large, and the total cost for compounding a locomotive may, for the purpose of comparison, be taken at 250 dollars for the complete change.

The cost of maintenance of a compound will be greater in the cylinders and less in the boiler. The somewhat better and more uniform draft on the fire and the lower rate of combustion in the firebox decreases the wear and tear on the furnace plates and tubes. Particularly is this true for such locomotives as have small boilers, and which work on heavy grades, especially in those sections of the country where fuel is cheap, for it is there that the fires are more recklessly handled, less attention is paid to fuel economy, more fuel is burned on the grate in a given time, and firebox failures are most frequent, particularly if the water is bad.

In this country locomotive boiler fires are forced more than the fires in any other type, except steam fire engine and torpedo boat boilers, and therefore the saving of coal due to compounding under average conditions, say 15 per cent., reduces the forcing of the fires so considerably that the effect is felt at once by the fireman. A boiler that may be difficult to fire for a single expansion locomotive may be easily handled for a compound doing an equal amount of work in the same time. Where the feed water contains much sediment or scale-producing salts, the reduction in the forcing of the boiler accompanying the use of compound cylinders is a decided advantage, and one that makes compounding worthy of consideration even where the fuel cost is a small part of the total cost of train expenses.

The maintenance of the cylinders, pistons, crossheads, guides, steam valves, steam chests, steam pipes and such other parts as are connected to the cylinders, and which are generally affected by compounding, amounts to about

3 per cent. of the total cost of locomotive repairs. The majority of all locomotive repairs are generally those which arise from the boiler, and a small saving in boiler repairs will more than offset the total cylinder repairs. If the compound system increases the cylinder repairs 100 per cent , the total cost is small, and will be offset fully in some cases by the consequent saving in boiler repairs. The total cost of repairs to a locomotive is not far from 1,200 dollars per year for large sizes. If the additional repairs to cylinders, etc., due to compounding, is as much as 100 per cent. for the parts affected, the additional cost, no allowance being made for the saving in boiler repairs, is but 36 dollars per year, and if there is any saving due to compounding that is worthy of the name, it would amount in money to more than 36 dollars in a single month, even with coal at $1.50 per ton. See Fig. 148.

147. Methods of Operating to Gain Economy.—It has been claimed that it is as economical to work a compound engine at $\frac{6}{10}$ cut-off in the h. p. cylinder, and wire-draw the steam through the throttle, for all grades of work requiring less power than $\frac{6}{10}$ cut-off, as it is to cut-off earlier in the h. p. cylinder. The fallacy of this for slow speed compound engines with good valve gears, or for high speed engines with adequate port and steam passages, and a suitable valve motion, is perhaps indicated by Fig. 45, which shows the losses due to wire-drawing in any engine, compound or single expansion, resulting from the loss in potential of steam pressure. It may be that in an inferior compound, where the steam passages and valve motion and ports are such as to give a very bad steam distribution at high speeds and short cut-offs, it would be more economical to use a longer cut-off and wire-draw the steam through the throttle, certainly it has been shown that the use of a long cut-off and a partially closed throttle gives more power at high speed, and the highest speeds so far attained by compounds have only been accomplished by this plan. But this is quite another matter, and has to do only with *power*, not with *economy*. An engine may not be running most economically at high speeds when it is generating the most cylinder power. So far as there is any evidence at all in the matter, everything goes to show that it is more economical to run with an open throttle at all times, when the boiler is not priming, than it is to wire-draw through the throttle. Probably the most economical compound, all other things being equal, is one that will

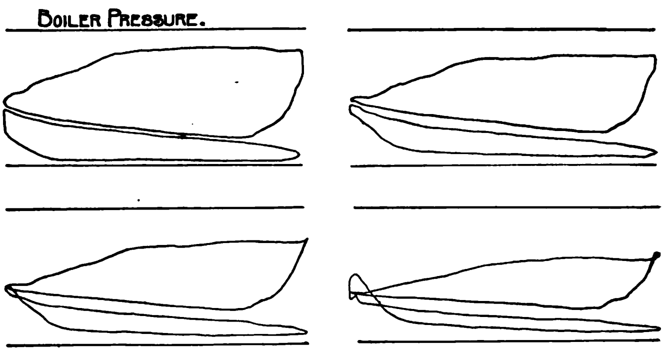


FIG. 149.
Indicator Cards Showing Steam Use When the Power is Regulated by the Throttle Lever.

run with sufficient power with a wide open throttle at cut-offs as early as $\frac{4}{10}$ of the stroke of the h. p. cylinder when at high speed. A compound locomotive should be designed, if possible, so that the power at all times can be regulated by the reverse lever, and the throttle be kept wide open.

Figs. 11 and 12, Indicator Cards Nos. 1 to 14, show one method of running compound locomotives; namely, by changing the point of cut-off as the speed increases. The effect of this in the matter of wire-drawing and compression at short cut-offs is clearly shown. Table A gives the data for these cards. Another method, and one which some compound locomotive builders have advised, is that shown by Indicator Cards Nos. 1 to 4, Fig. 149. The data for these cards is given in Table F F. This plan is one where the cut-off is not made less than about $\frac{6}{10}$ of the stroke but the power is regulated by the throttle.

TABLE F F.

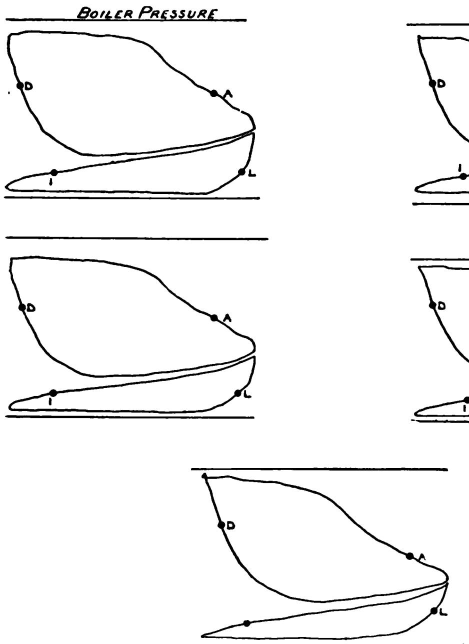
Card Number.	Boiler Pressure.	Revolutions per Minute.	Reverse Lever Notch.	Throttle opening.
1	155	144	4	7/8
2	160	228	4	7/8
3	160	246	5	1/2
4	150	308	5	1/4

When a compound engine is well proportioned for the work it is doing, the indicator cards at average speed compare favorably with those from a high speed stationary compound engine. This is shown by Cards Nos. 1 to 5, Fig. 150, which were taken from a Baldwin ten-wheel compound passenger engine on the Erie Railroad. The data regarding these cards is given in Table G G.

TABLE G G.

Card Number.	Boiler Pressure.	Revolutions per minute.	Speed. Miles an Hour.
1	180	120	25.71
2	180	160	34.28
3	180	160	34.20
4	163	140	29.80
5	179	172	36.85

This question of the proper method of operating has two sides to it, the economical, taking into consideration *only the fuel used;* from this standpoint it is better to run with a full throttle and vary the power by changing the point of cut-off. The other point of view takes into consideration *only the capacity* of the locomotive to haul trains at high speeds. Viewed from this last standpoint it is better with compound locomotives at high speed not to regulate the power entirely by the reverse lever, but to use a rather long cut-off and run by the throttle. A long cut-off gives larger port openings and a later exhaust closure. This reduces the wire-drawing through the valve and decreases the compres-



FIG, 150.
Indicator Cards From Vauclain Compound, Showing Steam Distribution at Low Speed.

sion. Compound locomotives have more power at high speeds when run in this way. As has been shown before, the loss of potential of pressure by closing the throttle carries with it a loss of efficiency that is not made up by the gain in the saving of cylinder condensation by the superheating that comes from wire-drawing through the throttle. See Fig. 45. If a compound locomotive can be run entirely by the reverse lever it is better to do so, but if the valve motion is such as to cause excessive compression at high speed there is only one way to get a substantial cylinder power, and that is by using a long cut-off and wire-drawing steam through the throttle to reduce the amount used per stroke to a point where the boiler can keep up the supply. If both a full throttle and a long cut-off are used at high speed, the vast amount of steam used would quickly drain the boiler and without a really useful result in the way of an increase of speed; this is for the reason that under such conditions the increase of back pressure in the cylinders, owing to the resistance of the exhaust nozzles, to a considerable extent offsets the greater forward pressure on the piston.

Those who have selected the tandem in preference to the non-tandem four-cylinder type, have done so with the expectation of gaining a mechanical construction of pistons and crossheads that is more theoretically perfect, 120, 123. With the tandem type, the annular piston of the Johnstone type and the uneven pressure on the crosshead of the Vauclain type are avoided; but, on the other hand, more parts are added and the front cylinder of the tandem is placed where it is more liable to damage in minor wrecks and collisions. It makes the front truck less accessible and interferes with some kinds of snow-plow and flanger attachments. The same number of piston rod stuffing-boxes are generally used with the tandem as with the Johnstone; the Vauclain has one less. The intermediate stuffing-boxes for tandems are generally a problem difficult to solve, 127. If enough room is taken to make them readily accessible, the over-all dimensions of the cylinders, lengthwise of the engines is greater than is desirable, 124-125. Hence, designers have been led to attempt to combine the stuffing-boxes between the h. p. and l. p. cylinders, and special boxes of small lengthwise dimensions have been devised. Such combined stuffing-boxes as have been put in actual service are practically inaccessible without great labor and delay, 127, so they may be said to be impracticable, as bad leakages cannot be readily discovered. The Mallet design of cylinders and stuffing-boxes for tandems, 125, is probably the best that has been put in service, but perhaps the valve rod arrangement of the Hungarian tandems are better, 124. It must be said that the tandem type has greater disadvantages than the two-cylinder receiver type in point of mechanical construction, and will probably be more inconvenient in a railroad shop and cost more to keep in repair than the Vauclain or Johnstone, as there are more parts to care for and they are not so easy of access.

Those who have selected the tandem types have done steam efficiency, and no advantages in practical operation for these types that are not possessed by simpler types, they may be dropped from further consideration, 133–134.

149. **Three-Cylinder Three-Crank Types.**—This type is open to the same objections as the four-crank four-cylinder type, and has the same disadvantages and is not superior to the simpler types in any particular. This type is, however, simpler in construction than the four-cylinder four-crank type, 128–132. All that has been said in the preceding about the four-cylinder four-crank types applies with equal force to the three-cylinder three-crank types, 148.

150. **Four-Cylinder Tandem Two-Crank Types.**—This type is made with and without a receiver, and can be operated by one valve for each pair of cylinders, like the Du Bousquet (Woolf) tandem engines on the Northern Railways of France, 118, or with two valves, as with the Mallet, Hungarian and Brooks Locomotive Works designs, 124–127. Only two sets of guides, crossheads, connecting rods and link motion are necessary for this type, the same as with the ordinary single expansion engine. So far as the theoretical economy or the starting power is concerned, it matters not with this type whether a receiver be or be not used, or whether the steam be controlled with two valves, or one for each pair of cylinders. The receiver designs give, perhaps, the best steam distribution, and the use of two steam valves gives better steam port openings and larger and straighter steam passages than can be obtained with one valve, and therefore for high speeds the receiver type with two steam valves is better, as the steam distribution is more readily controlled and variations can be made with less changes in details, 73–76.

The theoretical efficiency of this type is practically identical with that of other four-cylinder compounds, and is nearly the same as that of the Vauclain and Johnstone types, which are four cylinder two-crank non-tandem compounds, 151.

Those who have selected the tandem in preference to the non-tandem four-cylinder type, have done so with the expectation of gaining a mechanical construction of pistons and crossheads that is more theoretically perfect, 120, 123. With the tandem type, the annular piston of the Johnstone type and the uneven pressure on the crosshead of the Vauclain type are avoided; but, on the other hand, more parts are added and the front cylinder of the tandem is placed where it is more liable to damage in minor wrecks and collisions. It makes the front truck less accessible and interferes with some kinds of snow-plow and flanger attachments. The same number of piston rod stuffing-boxes are generally used with the tandem as with the Johnstone; the Vauclain has one less. The intermediate stuffing-boxes for tandems are generally a problem difficult to solve, 127. If enough room is taken to make them readily accessible, the over-all dimensions of the cylinders, lengthwise of the engines is greater than is desirable, 124-125. Hence, designers have been led to attempt to combine the stuffing-boxes between the h. p. and l. p. cylinders, and special boxes of small lengthwise dimensions have been devised. Such combined stuffing-boxes as have been put in actual service are practically inaccessible without great labor and delay, 127, so they may be said to be impracticable, as bad leakages cannot be readily discovered. The Mallet design of cylinders and stuffing-boxes for tandems, 125, is probably the best that has been put in service, but perhaps the valve rod arrangement of the Hungarian tandems are better, 124. It must be said that the tandem type has greater disadvantages than the two-cylinder receiver type in point of mechanical construction, and will probably be more inconvenient in a railroad shop and cost more to keep in repair than the Vauclain or Johnstone, as there are more parts to care for and they are not so easy of access.

Those who have selected the tandem types have done so because of some unusual conditions or some special service. Two at least of the tandems that have been built were made for the purpose of experiment.

151. **Four-Cylinder Non-Tandem Two-Crank Types, With and Without Receivers.**—None of this type with receivers have been built. Practically this class is represented solely by the Vauclain and Johnstone types, 120, 123. Many locomotives of the Vauclain type have been in service, and the results of practical trials are numerous. A number of the Johnstone type have been put into service on the Mexican Central Railway, and so far as can be learned no practical difficulties have been encountered. The theoretical efficiencies of these types are identical. The Johnstone has been used under exceptionally favorable conditions where speeds are slow and fuel is high in price, and a very great money saving has been gained.

The Vauclain has been used under all common conditions and in some very unusual classes of services, and the results have been correspondingly varied. In cases where the conditions have been the same as those under which the Johnstone has been used, the savings have been equally great, and in other cases very unfavorable conditions have led to little or no saving. With these compounds, as with all others the conditions control the saving in cost of fuel.

So far as the mechanical construction is concerned, the Vauclain has the advantage of greater simplicity and has parts that are more familiar to the average workman, 120–123. The uneven pressure on the crosshead of this type, 121, led many at first to expect trouble from actual service, but it has been pretty clearly demonstrated that so far as the crosshead and piston construction is concerned this type can be made to give as good service as an ordinary single expansion engine; but more care is required in design, better selection must be made of piston rod material, and the guides must be kept well lined up to prevent as much as possible the rocking motion that will always be induced by this construction. The use of large diameters for the h. p. piston rod has led to much criticism, but this has been the result of a too rigid piston rod connection to the crosshead. The large rods were used to remove the breakages of piston rods at the crosshead end that were so common in early designs. It is not believed that these large rods are necessary; in fact it is argued with some reason that smaller and more flexible (perhaps longer) piston rods would be less liable to break than the larger ones. However, the piston rod troubles are not greater with the Vauclain type than with the Laird crosshead type of single expansion engines at the present time, and are not such as to cause apprehension.

So far, the Vauclain type has been used with a single piston valve for controlling the steam in both cylinders, 120, and the results have been very satisfactory in slow service, but it has not been shown yet that a satisfactory steam distribution can be obtained at high piston speed with a short cut-off ($\frac{4}{10}$ of the stroke). See Figs. 11 and 12. The builders of this engine have advised a long cut-off ($\frac{6}{10}$ of the stroke) and a regulation of the power by the throttle at high piston speed, 147.

Also for the flat slide valve arrangement used with the Johnstone type; it has not been shown that the steam distribution is good at high speed when the cut-off in the h. p. cylinder is less than half stroke.

It is not as economical to use the throttle for regulation at any speed as it is to change the point of cut-off, and probably some change in the valve dimensions ordinarily used for these types will be needed to gain the maximum economy for high speeds.

In extreme width laterally over the cylinders, the Vauclain has the advantage, and this is an important point with some conditions. To the practical mechanic, the Vauclain has the simpler construction, and those who have charge of repairing locomotives will be of the opinion that the Vauclain has fewer parts to watch, and it is probable that with it the piston leakage and leakage from h. p. to l. p. cylinder will be less in actual service. On the other hand, the valve repairs must be less on the Johnstone, as the parts are simpler and are of ordinary form. The Vauclain valve bushing, 120, is not a simple detail and cannot be renewed without considerable expense; however, it wears but slowly when the valves receive proper care and suitable oiling.

Of all the four-cylinder types so far built, the Vauclain appears to be the most practical and easiest to keep in repair, at least it has been shown that the total additional cost of cylinder repairs for this type is too small a factor to be taken into account in a consideration of the value of compounding where the cost of fuel is of any considerable consequence.

In the past, those who have chosen the Vauclain type have generally done so because of its greater starting power and hauling power on inclines rather than from any superiority in theoretical efficiency. The Baldwin Locomotive Works have chosen it on account of its wide application to all classes of service. This results from the small over-all dimensions and from the fact that it is a type of compound that will do, under all conditions, all that a single expansion engine will do. In point of theoretical efficiency it is not equal to the two-cylinder type with receiver, as it has more cooling surface and a greater number of cylinders. In general, where other conditions are the same, and there is the same degree of expansion of steam, engines with the least number of cylinders will be most economical, and there will be less cylinder condensation.

The two-cylinder receiver type, with an independent exhaust for the h. p. cylinder at starting, is the strong competitor of the Vauclain type, as it has sufficient starting power and somewhat better theoretical economy, and has been shown to give better steam distribution at high speeds with short cut-off. See Figs. 11 and 15. No tests have been made that conclusively show the two-cylinder type to be more economical, although the theory of steam use would point that way. Yet it must not be forgotten that, taking into consideration only those two-cylinder and Vauclain compounds that have so far been built in this country, the Vauclain in most cases probably has the advantage at low speeds and with heavy work, as the l. p. cylinder capacity has been made much larger and greater expansion is had under equal conditions. With equal l. p. cylinder capacity the two-cylinder compound will have much greater over-all dimensions, and for large engines this is a point of weakness in the design that emphasizes the universal adaptability of the Vauclain type. With a double l. p. cylinder, as has been proposed by Mallet (see Fig. 100), and later by Lapage (see Fig. 28), for the nominally two-cylinder type, the needed large l. p. cylinder capacity can be obtained, but this affects the theoretical efficiency somewhat as it gives larger cooling surfaces and one more cylinder. All this goes to show the need of some accurate experiment at this time to determine the comparative efficiency of compounds. However, it is quite certain that where coal is expensive or the conditions severe a considerable saving in fuel cost will result from the use of compounds of any type, providing the designs of the details are correct and the proportions properly chosen, 145.

152. **Two-Cylinder Two-Crank Receiver Types.**— The theoretical efficiency of this type is greater than any of the others mentioned here. This arises from the less number of cylinders, less cooling surface, better arrangement of steam passages to prevent loss by radiation, possibility of reheating in the receiver, 54, and the more

complete control of cut-off and compression, 73-81. In practical service no superiority has been shown for this type in this country, as the advantages which it possesses have not been utilized. To get the necessary l. p. cylinder capacity greater over-all width has been thought to be necessary, and the tendency has been to keep the cylinders, both h. p. and l. p., smaller than they should be. This appears from Table C C, Appendix R.

In some cases it has already been found necessary to move the frames inward to get room for the cylinders. If the double l. p. cylinder be used, see Figs. 28 and 100, the same proportion of cylinder capacity to the work to be done can be obtained as has been used for two-cylinder compounds in Europe, and for four-cylinder compounds in this country.

In this type there is every chance to make a good insulation for all of the steam passages, and there is no excuse for placing h. p. steam on one side of a ½-inch wall and l. p. steam, or the cooler atmosphere, on the other side, as is generally done. This possibility of better heat insulation has not been utilized, except in the case of the Old Colony compound. See Figs. 72-75. The amount of re-heating in the receiver will vary with the temperature of the smoke box and the speed of the engine. By using a large copper receiver with a volume not less than three times that of the h. p. cylinder such re-heating as it is practicable to gain may be had. As the re-heating in the receiver is done by the waste heat in the furnace gases, all the re-heating that takes place is clear gain, and in this way it differs from re-heating by boiler steam, 54.

As the cut-off in the cylinders of this type can be readily varied there is a better chance to adjust the valves, 73-81, for the average conditions of operation than is the case with four-cylinder engines, with one valve for two cylinders and this more complete control of the distribution of steam, if taken advantage of, will give to this type greater efficiency at high speeds.

153. **In General about a Selection of a Suitable Design.**—The designer is confronted, so far as economy is concerned, with but practically two types, viz., the two-cylinder two-crank compounds with receiver, and the four-cylinder two-crank compounds with and without receiver. To the first belong the compounds of Mallet, von Borries, Worsdell, Lindner, Gölsdorf (Austrian), Schenectady, Rhode Island, Dean, Brooks two-cylinder, Richmond, Chicago, Burlington & Quincy Railroad, Pennsylvania Railroad, Rogers, Cooke and Pittsburgh compounds. To the second class belong the Vauclain, Johnstone, Brooks Tandem, Mallet Tandem, Hungarian Tandem, and the Du Bousquet types. So far as practical experience with compound locomotives goes, there is no evidence to show that any one of these types has the advantage of the others in point of economy, but there are many points of claimed advantage for each. All that is certain is, that the four-cylinder type is better in starting trains, and does more satisfactory work on grades than the two-cylinder type with automatic starting gear, but is not superior in this respect to the two cylinder type with a separate exhaust for the h. p. cylinder.

The four-cylinder type will start trains satisfactorily, and the starting gear needs but little consideration. What needs most attention is the mechanical construction of the driving mechanism and the arrangement of the valve motion and steam distribution.

With the two-cylinder compound the starting power needs first consideration. The mechanical construction is generally good. The valve motion and steam distribution can be readily made satisfactory by attention to the well-known principles of designing steam ports and passages, and the selection of a proper valve travel, and thereafter adjusting the cut-offs to suit the average working of the engine. This adjustment may be made in several ways, 73–81.

At the present time only two types of compounds have seen sufficient service in this country to prove their practicability. These are the two-cylinder two-crank receiver types, and the four-cylinder non-tandem two-crank receiver type, or, more concisely, the two-cylinder compound, and the Vauclain and Johnstone compounds.

It has been shown that the two and four-cylinder two-crank outside connected locomotives are perfectly practical machines, so far as steam use and hauling of trains is concerned, and therefore there is no advantage in the introduction of a third or fourth crank, with the consequent complication of parts.

It is not apparent that either the three or four-crank types will ever come into general use, for the reason that simplicity of design is of first importance in locomotive construction, and so long as locomotives with two cranks can be operated practically with such excellent results as at the present time (the theoretical saving being even greater with the two-crank than with the three or four-crank types) there will be little chance for a general introduction of compounds with more than two cranks. The single exception to this statement is in the case of double-bogie locomotives, in which there are necessarily four cranks per engine; such engines are in fact but two engines combined, each of which is a two-crank compound, and all the remarks that have been made here regarding the two-crank compounds apply with equal force to the double bogie.

It is a pertinent fact that while the three-cylinder three-crank compound has been given every advantage and possibility of success, and every opportunity to show its practical value, both in England on the Northwestern Railroad, and in France on the Northern Railways, and in this country on the Pennsylvania Railroad, yet no material advance has been made in the introduction of this type; while during the same period the use of the two-cylinder compound in England has been largely increased, and the four-cylinder two-crank non-receiver type has been given preference to all other types on the Northern Railways of France.

All further consideration of locomotives having crank shafts may be dropped, for the reason that they have no real nor apparent theoretical or practical advantages.

The question of hauling trains in case of accident to machinery is an important one from an operating standpoint. Compounds with h. p. and l. p. cylinders on the same side can haul trains with one side disconnected. With a separate exhaust for the h. p. cylinder and sufficient steam supply direct to the l. p. cylinder, the two-cylinder type can always haul trains when only one side is disabled. Without separate exhaust for the h. p. cylinder this type is practically helpless in case of a broken l. p. steam chest, or if there is a too small steam supply to the l. p. cylinder and the h. p. steam chest is broken. Where the cylinders and steam chest remain intact on both sides, this type can haul a train at considerable speed without an exhaust from h. p. cylinder when either side is disconnected.

So far as the details of the cylinders of compound locomotives are concerned, other than those that are referred to in these pages, including heat insulation, the best that can be done is to follow compound stationary engine practice, making due allowance for the wide variation of power demanded in locomotive operation, more particularly in starting trains quickly from a period of rest, and in hauling up short, steep inclines on otherwise level roads.

right angles with the h. p., and therefore either directly opposite each other or parallel. Assuming an equal division of work between the three cylinders, the most uniform turning moment will be obtained by placing the cranks at angles of 120 degrees with each other, but the difference in the distribution in the two l. p. cylinders will still exist.

An examination of the crank positions for the form of three-cylinder engine having two h. p. cylinders and one l. p. cylinder shows similar peculiarities in the distri-

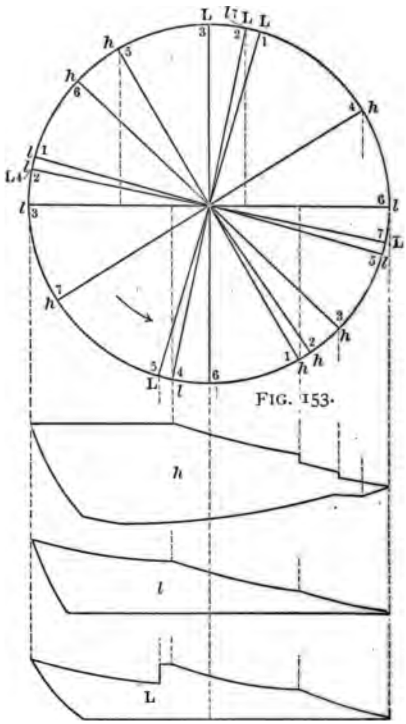


FIG. 154.
Elementary Indicator Cards from Three-Cylinder Compound.

bution. This will be evident from Figs. 155 and 156, which are lettered similarly to Figs. 151 and 153, *H* and *h* representing the two h. p. cranks, which are at right angles, and *l* the l. p. crank, which makes angles of 135 degrees with the others. The distribution in Fig. 155 is the same as that in Fig. 151, and that in Fig. 156 is the same as that in Fig. 153. It will be seen that there is readmission to the l. p. cylinder in both figures; but at the earlier cut-off of $\frac{4}{10}$ it is not probable that the effect on a l. p. indicator card would be noticeable. Placing the cranks at angles of 120 degrees would, as in the first arrangement of cylinders, produce very little change in the indicator cards.

APPENDIX.

NOTE.—In the following,

v = volume of h. p. cylinders in cubic inches.
V= " " l. p. " " " "
C= " " receiver " " "
R= ratio " volumes of h. p. and l. p. cylinders.

A. Example of Calculation for Mean Effective Pressure during one Stroke, 7.—Let the gauge pressure at the point b, Fig. 1, be 145.3 pounds per square inch, so that the absolute pressure will be 160 pounds. As cut-off takes place at one-half stroke the ratio of expansion $r = 2$, and therefore the final pressure in the h. p. cylinder will be one-half of 160 = 80 pounds. From the tables previously referred to we find that for $r = 2, p_m = .847\ p_1$. and therefore $p_m = .847 \times 160 = 135.5$ pounds absolute pressure, which is the mean pressure between a and c measured from the zero line of pressure.

B. Example of Calculation for Mean Effective Pressure during Expansion, 7.—For example, the pressure at b, Fig. 3, is 160 pounds and the volume at c is twice that at b. The ratio of expansion is therefore 2, and by reference to a table of hyperbolic logarithms we find $p_m = 160\ \frac{.693}{2-1} = 160 \times .693 = 110.9$ pounds between b and c. This is for one-half of the stroke, and for the first half, from a to b, the mean pressure is 160 pounds, therefore the average for the whole stroke would be

$$\frac{160 + 110.9}{2} = 135.5.$$

C. Example of Calculation for Pressure in the Receiver, 21.—In the present case, Fig. 1, assume the capacity of the receiver to be equal to that of the h. p. cylinder, or $C = v$, and let the pressure at e be taken at 96 pounds. At d the steam fills the h. p. cylinder + receiver, and at e fills one-half the h. p. cylinder + receiver; therefore the compression is from $v + C = 2\,v$ to $.5\,v + C = 1.5\,v$, and the ratio of compression is $2\,v \div 1.5\,v = 1.33$. The pressure at d is then $96 \times .75 = 72$ pounds, and the mean pressure between e and d is $96 \times .86 = 82.6$ pounds. At f the volume occupied is that of one-half the l. p. cylinder + receiver. Assuming for the present case that the l. p. cylinder is 2.5 times the h. p. cylinder, or $R = 2.5$ the expansion will be from $1.5\,v$ to $\frac{2.5\,v}{2} + C = 2.25\,v$, or the ratio of expansion is $2.25\,v \div 1.5 = 1.5$. The pressure at f is then $96 \times .67 = 64$ pounds, and the mean pressure between e and f is $96 \times .81 = 77.8$ pounds.

D. Final Pressure; Total Expansion, 45–52.—In an elementary compound engine, a certain fraction of the h. p. cylinder is filled with steam from the boiler at each stroke, and after expanding in both cylinders this mass of steam finally fills the l. p. cylinder before it is exhausted into the atmosphere or condenser. For example, if the cylinder ratio is 2.5 and the h. p. cut-off is at one-half stroke, $.5\,v$ is the volume admitted from the boiler at each stroke, and this finally fills the volume $2.5\,v$ before it is exhausted.

The steam is therefore expanded to 5 times its initial volume, or the ratio of total expansion is 5, and the final pressure at which it is exhausted will be $\frac{1}{5}$ of the initial pressure, or 32 pounds in the case we have used for purposes of illustration. Similarly, if the h. p. cut-off was at $\frac{3}{8}$ stroke the ratio of total expansion would be $2.5 \times \frac{8}{3} = \frac{20}{3} = 6\frac{2}{3}$, and the final absolute pressure in the l. p. cylinder would be $\frac{3}{20}$ of $160 = 24$ pounds. It will be noted that the only data required in determining the total expansion and final pressure in an elementary engine are the ratio of the cylinders and the h. p. cut-off, or, in other words, these results are independent of the capacity of the receiver and of the l. p. cut-off. The effect of the size of the receiver is seen in the shape of the indicator cards due to the compressions and expansions; but how many and how large these variations are does not affect the final pressure. The office of the l. p. cut-off is to control the division of the work between the two cylinders. In a compound engine, which exhausts into the atmosphere, the steam can, under the best and most favorable conditions, be expanded economically until the boiler pressure is reduced to the atmospheric pressure. Steam at 160 pounds absolute could, therefore, be expanded $\frac{160}{14.7} = 11$ times, nearly.

E. Drop in Pressure in Receiver, 26.—Taking $R = 2.5$, $C = v$, h. p. cut-off at $\frac{1}{2}$ stroke, and l. p. cut-off at $\frac{1}{2}$ stroke, we have the final pressure at the end of the expansion in the l. p. cylinder equal to $\frac{1}{5}$ of 160, or 32 pounds. The ratio of expansion in the l. p. cylinder is 2, therefore the pressure at the point f is $32 \times 2 = 64$ pounds. Then, knowing the ratio of expansion from e to f, as already calculated to be 1.5, we have the pressure at $e = 64 \times 1.5 = 96$ pounds, which was assumed for the time in calculating the variations of pressure in the receiver. Working back from this still further, we find the pressure at d as before, 72 pounds, and as the pressure at c is 80 pounds, there has been a *drop* in pressure of 8 pounds when the h. p. exhaust opened When the l. p. steam valve closed at f, the pressure of the steam left in the receiver was 64 pounds. Then when the h. p. exhaust opened, the steam which filled the h. p. cylinder at a pressure of 80 pounds mixed with this, and gave a resulting pressure of 72 pounds. To prevent drop in an elementary engine, it is only necessary to adjust the cut-off of the l. p. cylinder so that the volume of steam drawn by it from the receiver equals that of the h. p. cylinder. For instance, with dimensions already given in this paragraph, it will be evident that when the l. p. cut-off is at $\frac{1}{2.5}$ or $\frac{2}{5}$ of the stroke, here will be no drop, because $\frac{2}{5}$ of the l. p. cylinder is equal to the whole h. p. cylinder in volume, and if we withdraw from the receiver at each stroke a volume which is equal to that received from the h. p. cylinder, the pressure in the receiver will not be reduced. This can also be readily shown by calculating back from the final pressure in the l. p. cylinder as before. Suppose $e\,f$ to represent $\frac{2}{5}$ of the l. p. stroke instead of $\frac{1}{2}$, as shown in the figure, then the pressure at f would be $32 \times \frac{5}{2} = 80$ pounds, which would be the pressure in the receiver when the h. p. exhaust opened; and as this is also the final pressure in the h. p. cylinder, there would be no drop. There is, of course, always more or less drop due to wire-drawing and friction in passages which cannot be prevented, and it must also be borne in mind that all of these calculations are based on the assumption that pressures vary inversely as the volumes.

F. Mean Effective Pressure; Equivalent Pressure in One Cylinder, 7.—With the data already used the mean forward pressure in the h. p. cylinder was found to be

135.5 pounds. The mean receiver pressure, or h. p. back pressure, is 80.2 pounds, and thus the mean effective pressure in the h. p. cylinder is $135.5 - 80.2 = 55.3$ pounds. For the l. p. card, the mean pressure between e and f was found to be 77.8 and the pressure at f was 64 pounds. The mean presssure between f and g is $64 \times .693 = 44.4$ pounds. The mean forward pressure for the stroke is then $\frac{77.8 + 44.4}{2} = 61.1$ pounds, and assuming a back pressure of 18 pounds, or 3.3 above the atmospheric pressure, the l. p. mean effective pressure will be $61.1 - 18 = 43.1$ pounds.

As the ratio between the cylinder areas is 2.5, assuming the stroke to be the same in both cylinders, as it generally would be in practice, one pound per square inch on the l. p. piston is equivalent to 2.5 pounds per square inch on the h. p. piston. We can thus readily find the effective pressure in a single cylinder, which is equivalent to the effective pressure in the two cylinders of the compound engine. Ordinarily the mean pressure is thus referred to the l. p. piston, although a reference to the h. p. piston is more convenient for some purposes. In the present case, the effective h. p. pressure referred to the l. p. piston is $\frac{55.3}{2.5} = 22.1$ The total effective pressure referred to the l. p. piston is then $22.1 + 43.1 = 65.2$ pounds. From this we find that the proportion of the total work which is done by each cylinder is, in h. p., $\frac{22.1}{65.2} = .34$, and in l. p. $\frac{43.1}{65.2} = .66$. If the pressures are referred to the h. p. piston, we have $43.1 \times 2.5 + 55.3 = 107.8 + 55.3$ 163.1 as the equivalent pressure in one cylinder of the same size as the h. p. cylinder. Formerly common practice was to make the h. p. cylinder of a compound locomotive of the same size as one cylinder of the single expansion engine which it is intended to replace. On this basis the theoretical compound engine under discussion would be developing the same work as the single expansion engine when the latter was developing a mean effective pressure of ½ of $163.1 = 81.6$ pounds in each cylinder.

G. Example of Calculation for Mean Effective Pressure when Clearance is taken into Account, 4.—As an example of the application of the formula, let the apparent cut-off be at ⅓ stroke with 8 per cent. clearance. The actual ratio of expansion is then $\frac{1 + .08}{.33 + .08} = 2.63$, and the mean pressure between b and c will be $p_1 \frac{.967}{1.63} = .594 p_1$. This is for ⅔ of the stroke, and for the first third the mean pressure equals p_1. The mean for the stroke is therefore $\frac{2 \times .594 p_1 + p_1}{3} = .73\, p_1$. The mean pressure calculated by formula without correction would be $.70\, p_1$.

H. Derivation of Formula for Tractive Force, 62.—The work done in the cylinders in inch-pounds is $2 \times$ area in square inches $\times$ mean effective pressure $\times$ twice the stroke in inches $= 2 \times \frac{1}{4} \pi d^2 \times p \times 2 s.$; that at the rim of the driving wheels is the pull in pounds $\times$ the circumference of the wheel in inches $= T \times \pi D$; therefore,

$$T = \frac{2 \times \frac{1}{4} \pi d^2 \times p \times 2 s}{\pi D} = \frac{d^2 p s.}{D}$$

l. Some further Discussion of Three-Cylinder Three-Crank Compounds, 129–134.—In the three-cylinder receiver type the ratio of the volumes of the cylinders can be made greater than is practicable with two cylinders, and by a proper arrangement of cranks a more uniform rotative power can be secured.

The two arrangements of three-cylinder compound engines that have been applied to locomotives are with one h. p. and two l. p. cylinders by the Northern Railway of France, with two h. p. cylinders and one l. p., the arrangement of the Webb type, 128–132.

Steam Distribution.—The fundamental theory of the elementary three-cylinder compound engines does not differ from that of two-cylinder compound engines. The only differences which exist are the result of the relative angles of the cranks, and are to be found in the variations in the turning moments and in the variations in pressures in the receivers. Each case must be individually analyzed, and the only difference between such analyses and those already given for two-cylinder engines is the greater complication which arises from having three cranks to consider instead of two. As an example of the method to be preferably followed in attempting such an investigation, an arrangement of cranks which has been used for a locomotive is selected. In this form the low-pressure cranks are at right angles and the high-pressure crank makes angles of 135 degrees with them. In the first place we assume the following data: In the high-pressure cylinder, cut-off, .75; release, 90; compression, .90; in the low-pressure cylinders the same distribution.

In Fig. 151 are shown successive positions of the three cranks, h representing the high-pressure crank, L one low-pressure crank, and l the other. Assuming the direction of the revolution to be as indicated by the arrow, an exhaust takes place from the high-pressure cylinder when its crank is at h_1. One low-pressure crank, l_1, is then just commencing a stroke, and the other L_1, has accomplished about .57 of a stroke, the effect of the angularity of the connecting rods being neglected. From these positions there is free communication between the three cylinders and the receiver until L moves to L_2, where the cut-off takes place in that cylinder, the other l. p. crank being then at l_2 and the h. p. crank at h_2. From these positions expansion continues in the cylinder L, while there is still free communication between the other l. p. cylinder, the receiver and the h. p. cylinder until the l. p. crank L arrives at L_3, when steam is again admitted to that cylinder for the return stroke. The other l. p. crank is then at l_3, and the h. p. crank is at h_3. All three cylinders are now again in communication, and remain so until the cut-off position l_4 is reached, the other cranks then being at L_4 and h_4. The two cylinders which are represented by h and L remain in communication until the positions numbered $_5$ are reached, when steam is again admitted to the cylinder l. Soon after this the h. p. exhaust takes place at h_6, and a fresh supply of steam is admitted to the receiver, from which it enters both l. p. cylinders whose cranks are at L_6 and l_6. These positions correspond to those numbered $_1$, the direction of the piston movement only being changed. It is clear that, when the exhaust takes place from the h. p. cylinder, the l. p. piston corresponding to l is always near the beginning of a stroke, while the other is near the middle of its stroke. The effects of this distribution in the l. p. cylinders are shown in Fig. 152 by indicator cards, which are constructed on the assumption of rapid valve movements and neglecting the irregularities which are caused by the connecting rods. The cards are not drawn to a scale and the variations in pressures are purposely exaggerated. With a relatively large receiver the drop in pressure at l_3 and L_5 will be very small. In practice the readmission at $_1$ would produce a hump in the card L, while the card l would have a form which would apparently indicate that the valve was late in opening.

At earlier points of cut-off somewhat different results will be found. These are illustrated by Figs. 153 and 154, in which it is assumed that cut-off takes place at .4 and release at .75 of the stroke in all three cylinders. Taking the direction of revolution as before, when release occurs in the h. p. cylinder at h_1, one l. p. crank is at L_1 and the other is at l_1. A very slight movement brings the crank L to its cut-off position L_2, soon after which steam is admitted to the other l. p. cylinder at l_3, and that cylinder is in communication with the receiver and the h. p. cylinder until its cut-off

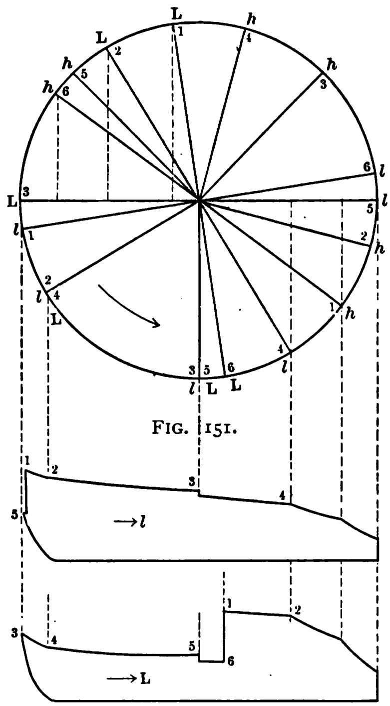


FIG. 151.

FIG. 152.

Elementary Indicator Cards from Three-Cylinder Compound.

point is reached at l_4. There will then be slight compression in the h. p. cylinder and the receiver until steam is admitted to the L cylinder at the beginning of its next stroke. The remaining events of the revolution are similar to those already noticed and will be made clear by a study of Fig. 153. It will be seen that there is still readmission to the l. p. cylinder L, but that this does not effect the form of the card from the other l. p. cylinder. With this arrangement of cranks and with the same valve adjustment the indicator cards from the two l. p. cylinders will be unlike for all points of cut-off. There is in fact but one arrangement of cranks for which the distribution in the l. p. cylinders will be the same, and that is when the l. p. cranks are both at

It is evident, from the preceding partial analysis of the steam distribution, that the construction of theoretical indicator cards for three-cylinder compound engines will be considerably more difficult than for the two-cylinder type, but that the same formulas and methods of construction can be used. The remarks which were made in discussing two-cylinder compound locomotives in regard to the effect of varying the capacity of the receiver and the results of changing the points of cut-off are equally applicable to three-cylinder engines. In fact, the only differences are those in the steam distribution, which have been already discussed, and which depend upon the angles made by the three cranks.

A mathematical discussion of the three-cylinder type of compound engine, having one h. p. cylinder and two l. p. cylinders, and with the cranks placed at angles of 120 degrees with each other, will be found in the appendix to "The Marine Steam Engine," by R. Sennett. The form having two h. p. cylinders and one l. p. cylinder does not appear to have been used in marine practice, and its use is not to be expected, inasmuch as one of the chief reasons for using three cylinders instead of two is to avoid excessively large l. p. cylinders.

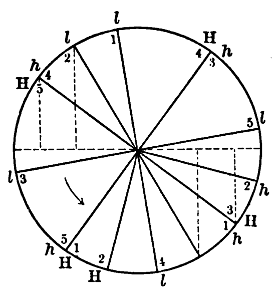


FIG. 155.

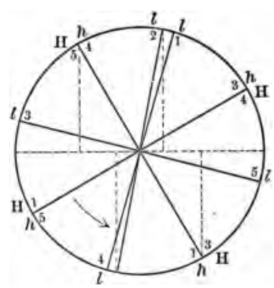


FIG. 156.

Crank Circles, Three-Cylinder Compound.

In attempting to determine the size of cylinders for three-cylinder compound locomotives, the best guide will undoubtedly be the results obtained with locomotives of that form in practice. When such information is not obtainable, the most satisfactory method will be that advocated under similar circumstances for two-cylinder compound engines, *i. e.*, the construction of, what were called for convenience, elementary indicator cards, and the alteration of these as experience dictates, to allow for wire-drawing during the opening and closing of valves; drop in pressure, etc. The proportions which appear to have been generally adopted by Mr. Webb are, h. p. cylinders, 14 inches in diameter; l. p. cylinder 30 inches in diameter; stroke of all pistons, 24 inches. The ratio of the volume of the l. p. cylinder to that of both h. p. cylinders is thus about 2. 3. Assuming a mean forward pressure of 175 pounds gauge, in the h. p. cylinders, and a back pressure in the l. p. cylinder of 3 pounds above the atmospheric pressure and an equal division of work, we can make an approximate estimate of the maximum power of the engine as follows: The area of the l. p. piston is 4.6 times that of one h. p. piston, and, if the work is to be the same in both, the mean pressure in a h. p. cylinder must be 4.6 times that in the l. p. cylinder. As the total range of pressure is 172 pounds, and as the mean receiver pressure is approximately the same as the mean h. p. back pressure and the mean l. p. forward pressure, we have: 4.6 × l. p. mean effective pressure = 172 — l. p. mean effective, whence l. p. mean effective = 172 ÷ 5.6 = 30.7 pounds. The mean receiver pressure is then 30.7 + 3 = 33.7 by gauge, and the mean effective in the h. p. cylinders is 175 — 33.7 = 141.3 pounds. A similar calculation can, of course, be made with any assumed mean forward pressure, and this method can also be used for making an approximate comparison of the maximum work done in the cylinders of the three-cylinder compound with that in ordinary locomotives. For example, if the mean forward pressure in the latter is 150 pounds and the back pressure is 3 pounds as before, the total effective pressure during a stroke will be 2 × 147 × area of one piston. To be the equivalent of the compound locomotive this must equal 3 × 141.3 × area of one h. p. piston. This gives in the present case 221.9 square inches as the piston area of the simple engine, or in other words a simple engine having two cylinders about 16.8 inch in diameter, would be equal in power, with the assumed pressures, to the compound engine having cylinders 14, 14 and 30 inches in diameter, the stroke being the same in all cylinders.

The same method can be used to find dimensions for an equivalent three-cylinder engine having one h. p. and two l. p. cylinders. If the ratio of the volumes of the two l. p. cylinders to that of the h. p. cylinder is 2.3, each l. p. cylinder will be 1.15 times as large as the h. p. Therefore 1.15 × l. p. mean effective pressure = 172 — l. p. mean effective, whence l. p. mean effective = 80 pounds. The mean receiver pressure will be 83 pounds gauge, and the h. p. mean effective pressure will be 175 — 83 = 92 pounds. To find the piston areas we have 92 × area of the h. p. piston for this engine = 141.3 × area of a 14-inch cylinder, which gives an area of 236.3 square inches, 17.35 diameter, for the h. p. piston, and 1.15 times this or 271.8 square inches, 18.6 diameter, for each l. p. piston. An engine having one h. p. cylinder 17.35 inches in diameter and two l. p. cylinders 18.6 inches in diameter, is thus equivalent with the assumed pressures to one having two h. p. cylinders 14 inches in diameter and one l. p. cylinder 30 inches in diameter. The distribution of work among the three cylinders is considered in what follows.

Distribution of Work.—It was shown in the theoretical discussion of the distribution of work between the cylinders of two-cylinder receiver compound locomotives, that with the same points of cut-off in both cylinders and with the ratios of cylinder volumes which are practicable in locomotives, considerably more than one-half of the total work will be done by the l. p. cylinder. It was also demonstrated that the work can be to a great extent equalized by making the cut-off in the h. p. earlier than that in the l. p. cylinder.

The same process of reasoning can be applied to the three-cylinder type of compound engines, inasmuch as we may regard this form as a development of the two-cylinder type, produced by substituting either two smaller h. p. cylinders for the original h. p. cylinder, or else two smaller l. p. cylinders for the original single l. p. cylinder. It is, therefore, to be expected that, with the same points of cut-off in all three cylinders, considerably more than one-half of the total work will be done in the single l. p. cylinder of the Webb type of compound locomotive, and in the two l. p. cylinders of the other form of three-cylinder compound locomotive which, for the sake of brevity, may be called the French type. We may even go a step further and say that, with the ratios of cylinder volumes which are practicable, the total work cannot be so divided that much less than one-half of it will be done in the l. p. cylinders. This statement is borne out by the published indicator cards of the Webb locomotive and leads to some interesting conclusions.

These indicator cards show that the proportion of the total work which is done in the l. p. cylinder is from 50 to 65 per cent. at various speeds, with the l. p. valve in full gear. As making the l. p. cut-off earlier would increase the proportion of work done in that cylinder, it follows directly that the l. p. cylinder's share of the total work is *at least* 50 per cent. As the Webb locomotive has no coupling rods between the h. p. and l. p. axles, and as the weight on each pair of drivers is very nearly the same, it is evident that this division of work is the best under the circumstances. This point and others can be well illustrated by a diagram of crank efforts. Such a diagram is shown by Fig. 157, which was constructed from indicator cards of a Webb locomotive. Steam was cut off at about ten inches in the h. p. cylinders, and the l. p. admission

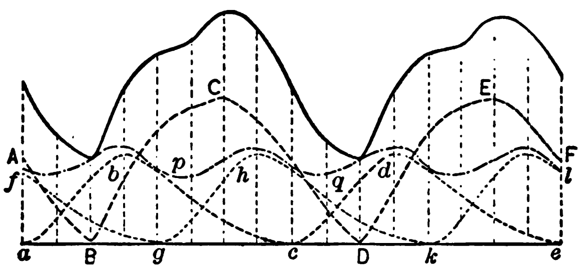


FIG. 157.

Diagram of Turning Moments, Three-Cylinder Compound.

FIG. 158.

Diagram of Turning Moments, Three-Cylinder Compound.

was at "full gear." The speed is not recorded, but from the form of the h. p. admission line is evidently not great. The mean pressure is approximately 81 pounds in the h. p. cylinders, and about 34 pounds in the l. p. cylinder, which is equivalent to $34 \times 2.3 = 78.2$ pounds in the two h. p. cylinders, the work done in the l. p. cylinder thus being nearly one-half of the total.

Referring to Fig. 157, *a b c d e* and *f g h k l* show the variations in the turning moments, or the tangential efforts on the cranks, of the two h. p. pistons, the cranks being at right angles, and the irregularity caused by the connecting rods being neglected. The combined efforts on these two cranks is shown by the curve *f p q l*. The variations in the turning moments on the l. p. crank are shown by a curve such as *B C D*, and if we assume that the l. p. crank makes angles of 135 degrees with the h. p. cranks, this curve and that for the other stroke *D E F A B* will be located as shown in the figure. Combining the h. p. and l. p. diagrams gives the full line curve in the figure which represents the variations in the pulling power of the locomotive during one revolution, as shown by the indicator cards, and therefore without taking the inertia of moving parts into consideration. A comparison of this full line curve with the curve of the combined h. p. cylinders *f p q l* shows that the angles between the l. p. and the two h. p. cranks are not of great importance. If the l. p. crank were moved back about 25 degrees, so that the maximum moment for the l. p. crank at *C* would coincide with the minimum for the combined h. p. cranks at *p*, the combined diagram for all three cranks would be somewhat more uniform, but the difference would not be great. A diagram of crank efforts on the assumption of uniform steam pressures throughout the stroke in each cylinder shows similar peculiarities. It has been suggested that this type of locomotive might be improved by placing the cranks

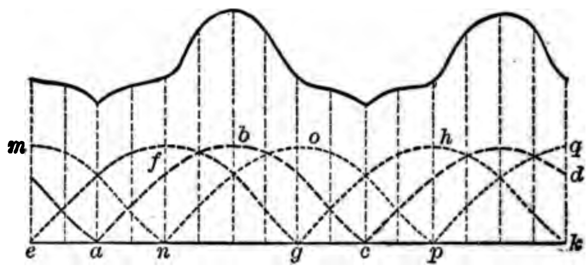


FIG. 159.
Diagram of Turning Moments, Three-Cylinder Compound.

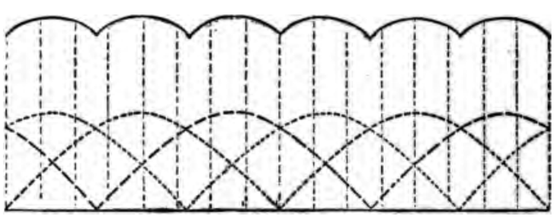


FIG. 160.
Diagram of Turning Moments, Three-Cylinder Compound.

at angles of 120 degrees and coupling the driving wheels. The effect of this, with the steam distribution and division of work used in the construction of Fig. 157, is shown by Fig. 158, in which the full line curve shows the variations in the combined rotative efforts on the three cranks. It will be seen that the minimum turning moment is greater than that in Fig. 157 and the maximum is less, so that there is a more uniform effort throughout the revolution. The performance of the locomotive at slow speeds would therefore be improved by this arrangement, but as the speed is increased the inertia of the moving parts tends to diminish this apparent advantage, so that it is doubtful if there would then be any practical gain by the introduction of coupling-rods.

Turning now to the French type of three-cylinder compound locomotives, it will be found that an application of the same method of reasoning leads to very different results. As has been pointed out, the steam distribution is different in the two l. p. cylinders, but it is nevertheless to be expected that more than one-half of the total work will be done in the l. p. cylinders with the same points of cut-off in all three cylinders. Also by adjusting the points of cut-off, the proportion of the total work done in the h. p. cylinder can be decreased. It is therefore possible with this type of engine to divide the total work equally among the three cylinders.

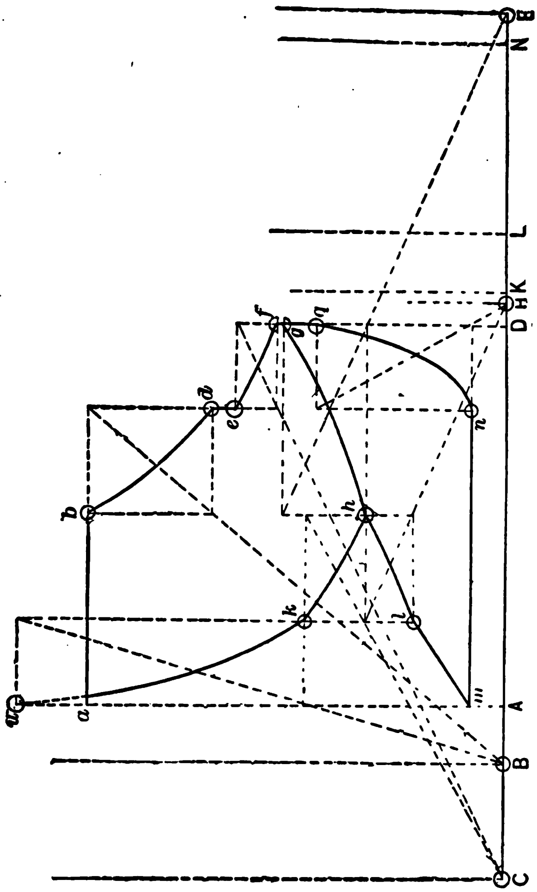


FIG. 161.
Diagram Showing a Method of Modifying Elementary Indicator Cards from Non-Receiver Compounds.

In this locomotive the two l. p. cranks are placed at right angles, and the h. p. crank is placed at 135 degrees with the others. A diagram of crank efforts with this crank arrangement and on the basis of an equal division of work, and steam admission during about ⅞ of the stroke, is shown by Fig. 159. In this figure, *a b c d* is the h. p. diagram, and *e f g h k* and *m n o p q* are the l. p. diagrams. The combined diagram for all three cranks is shown by the full-line curve. If the cranks were placed at angles of 120 degrees, the combined diagram would have the form shown by the full-line curve in Fig. 160, from which it is clear that this disposition of cranks would give a very constant turning moment.

J. Example of Modification of Elementary Indicator Cards to Approximate to Actual Cards for Non-Receiver Compounds, 3.—An example of indicator cards constructed in this way is given in Fig. 161, on a much smaller scale, however, than is advisable in practice. The assumed data in this case is as follows: Initial pressure, 175 pounds absolute; cylinder ratio, 3; l. p. back-pressure, 17 pounds absolute; cut-off in both cylinders, 0.5; release and compression in both cylinders, 0.78; volume of h. p. clearance, 15 per cent.; volume of l. p. clearance, 6 per cent.; volume of connecting passages, 0.3 of h. p. cylinder.. The scale of pressures used in the diagram is 80 pounds to the inch. For the benefit of those who may wish to construct such diagrams we will follow through this case in some detail.

The following symbols will be used:

v = volume swept by h. p. piston.
V = " " l. p. "
c = " of h. p. clearance.
C = " of l. p. "
i = " of intermediate or connecting passages.

The volumes occupied by the steam at the several lettered points on the diagram are, then,

At b, $= .5\,v + c = .65\,v$.
At d, $= .78\,v + c = .93\,v$.
At e, $= .93\,v + i = 1.23\,v$.
At f, $= v + c + i = 1.45\,v$.
At g, $= 1.45\,v + C = 1.63\,v$.
At h, before cut-off, $= .5\,v + c + i + C + .5\,V = 2.63\,v$.
At h, in l. p. after cut-off, $= .5\,V + C = .56\,V$.
At h, in h. p. and passages after cut-off in l. p., $= .5\,v + c + i = .95\,v$.
At k, before valve closure, $= .22\,v + c + i = .67\,v$.
At k, in h. p. after valve closure, $= .22\,v + c = .37\,v$.
At l, $= .78\,V + C = .84\,V$.
At n, $= .22\,V + C = .28\,V$.

The pressure at d and the curve between b and d may be found by constructing the curve through b with B as the origin, $A\,B$ being .15 of $A\,D$; or by calculation as the pressures may be taken inversely as the volumes, whence pressure at $d = 175 \times .65 + .93 = 122.3$ pounds. The drop in pressure from d to e depends upon the pressure at k, that in turn depends upon h, and so upon g. The pressure at g depends upon that at q and at f, and so upon e. In any case, there is but one pressure at h which will fulfil the conditions, and that pressure must be determined by calculation. Assuming for the moment that we know the pressure at e to be 112.5 pounds, the pressure at f will be $112.5 \times 1.23 + 1.45 = 95.4$ pounds. The pressure at g is determined by the mixture of the volume at f at 95.4 pounds with the volume of the l. p. clearance at pressure q. To find the latter we have pressure at $q = 17 \times .28 + .06 = 79.3$ pounds. Then pressure at g =

$$\frac{79.3 \times .18 + 95.4 \times 1.45}{.18 + 1.45} = 93.7 \text{ pounds.}$$

The pressure at $h = 93.7 \times 1.63 + 2.63 = 58.1$ pounds. The pressure at $k = 58.1 \times .95 + .67 = 82.3$ pounds. We can now find the pressure at e which is

$$\frac{122.3 \times .93 + 82.3 \times .3}{.93 + .3} = 112.5.$$

By combining these various expressions for pressures we can readily form a single equation from which the pressure at h can be calculated, which is, in fact, the method by which it was determined in this case.

Having found the pressures at e, g and h by calculation, the various curves of the diagrams can be readily constructed. For the curve between e and f a point C is used for the origin, which is found by laying off $B\ C$ equal to .3 of $A\ D$. The curve $h\ k$ is constructed from the same origin. The compression curve $k\ u$ is laid off from B. To find the origin for the curve $g\ h$, we proceed as follows: At g the steam occupies the volume $v + c + i + C$, and at h the volume occupied is $.5v + c + i + C + (.5V = 1.5v)$. The increase in volume is therefore equal to v, and therefore the scale of this part of the diagram must be such that the horizontal distance from g to h represents v, the volume of the h. p. cylinder. With this scale of volumes lay off $D\ K = .06V = .18v$, $K\ L = .3v$, $L\ N = v$ and $N\ E = .15v$; then E is the origin from which to construct the curve $g\ h$. For the curves $h\ l$ and $n\ q$ the origin is taken at H, which is found by laying off $D\ H = .06$ of $A\ D$, which for these curves represents the volume of the l. p. cylinder.

This diagram illustrates the difficulty of keeping the h. p. compression within reasonable limits.

K. Some Further Discussion of Four-Cylinder Receiver Compounds, 124–128.— The elementary theory of four-cylinder receiver compound locomotives is essentially the same as that of two-cylinder receiver engines, and the four-cylinder type may be regarded, as far as the cylinders are concerned, as formed from the two-cylinder type by substituting for each cylinder of the latter two cylinders having a joint volume equal to the corresponding single cylinder. It was shown in discussing two-cylinder receiver engines that, in making approximate calculations to determine proportions, the receiver pressure may be regarded as constant, assuming that the capacity of the receiver is large compared with that of a h. p. cylinder. It follows from this that the distribution of work in the cylinders is practically independent of the angle between the h. p. and l. p. cranks when a large receiver is used. If in a four-cylinder engine both h. p. cylinders exhaust into one receiver, which is the reservoir from which both l. p. cylinders are supplied with steam, the variations in pressure in this receiver during a revolution will presumably be less than in a two-cylinder engine.

We can, therefore, in the design of four-cylinder elementary engines, make use of formulas which are based upon a constant receiver pressure, proceeding at first as if the engine were to have but two cylinders. The formulas are those which are usually given for two-cylinder receiver engines, and are not of special value in the design of two-cylinder compound locomotives on account of the necessity of a very careful analysis of the steam distribution in that type of locomotive if the possible advantages of compound working are to be realized.

In subsequent formulas the letters have the following meaning:

v = volume of h. p. cylinder.
V = " " l. p. cylinder.
R = ratio of the cylinders, $V = R\,v$.
r = ratio of expansion in h. p. cylinder.
r' = ratio of expansion in l. p. cylinder.
p_1 = pressure in h. p. cylinder during admission.

p_2 = pressure in h. p. cylinder when exhaust opens.
p_3 = mean measure in the receiver.
" = pressure in l. p. cylinder during admission.
p_4 = mean l. p, back pressure.
All pressures are absolute pressures.

Neglecting the effects of clearance, the mean foi ard pressure in the h. p. cylinder is:

$$p_m = \frac{1 + hyp.\ log.\ r}{r} p_1$$

The mean effective pressure is $(p_m - p_3)$ and the work done in the h. p. cylinder during a stroke is $v\ (p_m - p_3)$. Similarly, the mean forward pressure in the l. p. cylinder is,

$$p'_m = \frac{1 + hyp.\ log.\ r'}{r'} p_3$$

the mean effective pressure is $(p'_m - p_4)$ and the work done in the l. p. cylinder during a stroke is $V\ (p'_m - p_4)$.

If the work is to be equally divided between the two cylinders, $v\ (p_m - p_3) = V\ (p'_m - p_4)$.

On the basis that volumes vary inversely as the pressures, we have,

$$p_2 = \frac{p_1}{r} = \frac{p_3}{r'}\frac{V}{v} = \frac{p_3 R}{r'}$$

By substituting the value for p_3 obtained from this equation in the preceding one, and reducing, the following is obtained:

$$\frac{p_1}{Rr}\left(hyp.\ log.\frac{r}{r'} - \frac{r'}{R}\right) + p_4 = O. \quad (4)$$

By means of this equation the ratios of expansion in each cylinder (r and r') for which the work done in each will be equal can be determined for any assumed values of p_1 p_4 and R. If it were required that there should be no drop in pressure at the end of the expansion in the h. p. cylinder, p_2 must equal p_3, from which it follows that r' must equal R. It will be found that equation (4) will give impossible values for r' for many values of r. As r becomes less or steam is admitted to the h. p. cylinder during a large part of the stroke, r' will be found to be less than one which is manifestly impossible, and shows that with a late cut-off in the h. p. cylinder the work cannot be equally divided between the cylinders. On the other hand, as r is made large, r' also increases until it is greater than R, which is an impracticable result, as the receiver pressure would then be higher than the pressure in the h. p. cylinder at the end of the expansion. For example, if we take $R = 2.3$, $p_1 = 190$ pounds absolute, $p_4 = 20$ pounds absolute, and $r = 1.33$, or cut-off at 0.75 of the stroke in the h. p. cylinder, the equation reduces to $hyp.\ log.\ r' + .4348r' = 0.3904$ from which $r' = 0.97$. As steam is admitted during the whole stroke when $r' = 1.0$, it is clear that with the above proportions more than one-half of the total work is necessarily done in the l. p. cylinder. If r is taken as equal to 4, with the other data the same as before, the value of r' will be found to be 3.75, or the l. p. cut-off would have to be placed at 1 ÷ 3.75 or 0.267 of the stroke. But as there will be no drop in pressure between the cylinders when $r' = R$, or when steam is cut-off in the l. p. cylinder at 1 ÷ 2.3=0.435 of the stroke, it follows that to equalize the work in the two cylinders at the earlier cut-off the receiver pressure would have to be higher than the pressure at the end of the expansion in the h. p. cylinder.

The engineers of the Paris, Lyons & Mediterranean Railway have applied a formula similar to the above in the determination of the proportions for a class of four-cylinder compound locomotives, and have shown the proper relations existing between the points of cut-off in the h. p. and l. p. cylinders graphically by a diagram similar to Fig. 162. This diagram is that given by Mr. C. Baudry, assistant engineer-in-chief of motive power and equipment. Formulas similar to the above will be found discussed at greater length in "Compound Engines," by Mr. Mallet. In Fig. 162 the horizontal distances represent the points of cut-off in the h. p. cylinder, and the vertical distances represent the points of cut-off in the l. p. cylinder. The inclined lines are curves which represent the solution of equation (4) for different values of R, the pressures used in the construction of the diagram probably being 213 and 21 pounds. For example, if $R = 2.5$ when the h. p. cut-off is at 0.4, the l. p. cut-off should be at about 0.5 in order to equalize the work. If the ratio $R = 2$, a cut-off at 0.4 in the h. p. requires a cut-off at about 0.58 in the l. p. cylinder. For the cases in which the equation gives values of r' which are too small, the cut-off for the l. p. cylinder is fixed

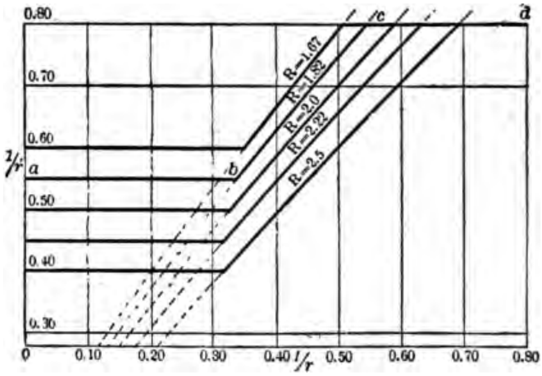


FIG. 162.

Diagram Showing the Ratios of Cut-off in High and Low-Pressure Cylinders of Four-Cylinder Compound.

at 0.8 or the maximum for full gear. For instance, taking $R = 2$, the l. p. cut-off would remain at 0.8, or full gear for all values of the h. p. cut-off greater than 0.58, although more than one-half of the work would be done in the l. p. cylinder. The other limit to the application of the formula is fixed by making the earliest l. p. cut-off that at which there will be no drop in pressure between the cylinders. So that finally, the relation between the points of cut-off in the two cylinders is shown by broken lines such as *a b c d*, for which $R = 1.82$. For example, if $R = 2$, the diagram shows that the points of cut-off should vary as follows:

High-pressure	.10	.20	.30	.40	.50	.60	.70	.80
Low-pressure	.50	.50	.50	.58	.70	.80	.80	.80

Three experimental types have been constructed by the Paris, Lyons & Mediterranean Railway. Fig. 163 shows the general arrangement of the compound locomotives intended for fast passenger service. The principal dimensions of these locomotives, and of the type of simple locomotive which formed the basis for the design, are given in Table JJ. The table shows that two compound locomotives of this and of each of the succeeding types have been built, which differ only in the number and diameter of the tubes. It will be seen that in the type of locomotive illustrated by Fig. 163 all four cylinders are placed beneath the smoke box, with their axes horizontal.

The two h. p. cylinders are between the frames and are connected to the forward driving axle. The l. p cylinders are connected to the rear driving axle. The axles are so coupled that the h. p. crank on each side leads the l. p. crank on the same side

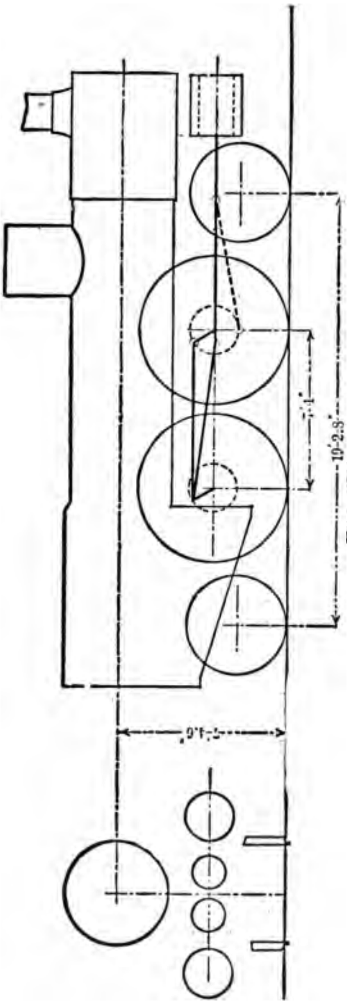


FIG. 163.
Four-Cylinder Compound Express Locomotive on Paris, Lyons and Mediterranean Railway.

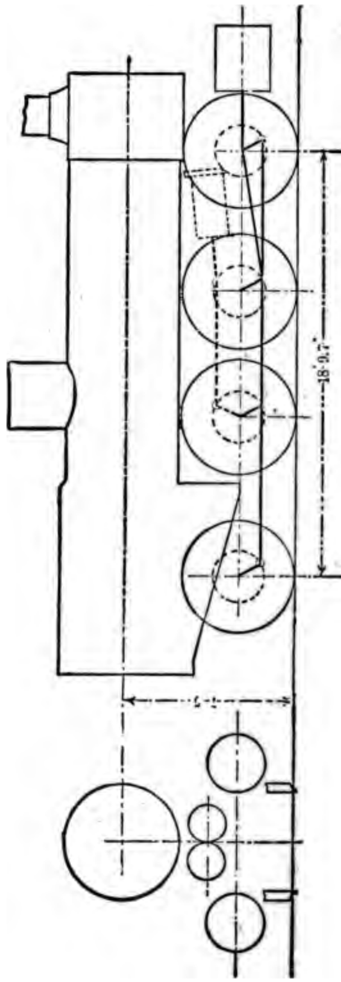


FIG. 164.
Four-Cylinder Compound Freight Locomotive on Paris, Lyons and Mediterranean Railway.

198°. The object of this arrangement is to obtain as large a value for the minimum starting power as possible. In Fig. 164 is shown the general arrangement of the four-cylinder compound locomotives for freight service. In this locomotive the second driving axle is connected to the l. p. cylinders, and the third axle to the h. p. cylinders. The h. p. crank on each side leads the l. p. crank 232° 48'. In the corresponding simple locomotive, of which the dimensions are given in the sixth column of the table, the rear axle is not a driving axle. Fig. 165 illustrates the arrangement adopted for locomotives for steep grades. The h. p. cylinders are connected to the second axle,

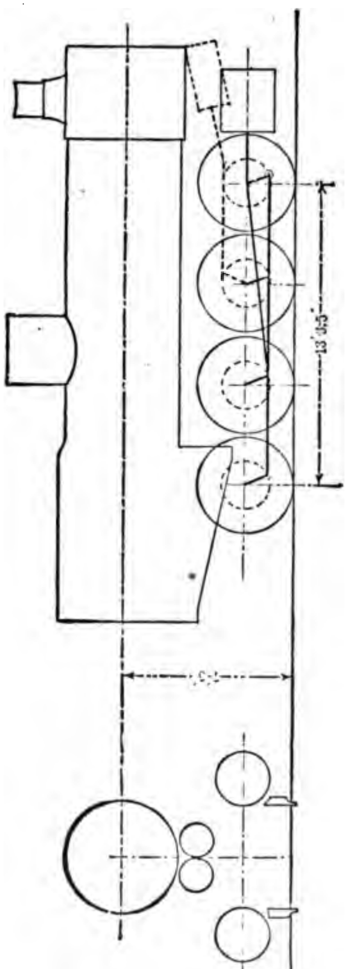


FIG. 165. Four-Cylinder Compound Freight Locomotive on Paris, Lyons and Mediterranean Railway.

and the l. p. cylinders to the third axle. The h. p. cranks lead the adjacent l. p. cranks, as in the other designs, but in this case the angle is 235° 54'.

The Walschaert valve gear is used for all of these locomotives, and the points of cut-off in the h. p. and l. p. cylinders are adjusted by means of a complicated cam arrangement, designed to fulfill the requirements indicated by Fig. 162. The starting gear adopted for these locomotives consists of simply an auxiliary steam pipe and cock for admitting steam from the boiler to the receiver, which is fitted with a safety valve as usual.

For the express locomotive, Fig. 163, in which the question of the balancing of the reciprocating parts at high speeds would be of most importance, the angle selected,

TABLE JJ.

Giving Dimensions of Paris, Lyons and Mediterranean Four-Cylinder Compound and Simple Locomotives.

	Fig. 163.			Fig. 164.			Fig. 165.		
	Compound.		Simple.	Compound.		Simple.	Compound.		Simple.
	H. P.	L. P.		H. P.	L. P.		H. P.	L. P.	
Number of cylinders	2	2	2	2	2	2	2	2	2
Diameter of cylinders, inches	12.21	19.68	19.68	13.38	21.26	21.26	14.17	21.26	21.26
Stroke of pistons, inches	24.41	24.41	24.41	25.59	25.59	25.59	25.59	25.59	25.98
Inclination of cylinders	0°0	0°	0°	7°48'	0°	0°	10°54'	0°	0°
Maximum travel of valve, inches	4.57	5.59	4.86	5.40	5.63	4.82	5.55	5.63	5.51
Outside lap of valve, inches, front	1.10	1.06	1.22	1.10	1.14	1.10	1.14	1.14	1.14
Outside lap of valve, inches, back	1.02	.95							
Inside lap of valves, inches	0	0	0.04	0.04	0.04	0.04	0.04	0.04	0.02
Maximum admission, per cent. of stroke	71.5	84.5	69.2	76.6	77.9	74.4	77.6	79.3	80.3
Steam ports, area, square inches	10.85	21.7	16.08	13.95	25.11	16.08	14.26	25.11	25.11
Steam ports, per cent. of piston area	9.35	7.13	5.35	10.13	7.09	4.59	9.04	7.09	7.09
	(1)	(2)		(1)	(2)		(1)	(2)	
Number of tubes	185	224	185	247	307	200	247	210	245
Diameter of tubes, outside, inches	1.97	1.77	1.97	1.77	1.58	1.97	1.97	2.17	1.97
Length between tube sheets, feet	13.24	13.24	16.23	14.28	14.28	16.49	13.62	13.62	17.57
Heating surface, fire-box, square feet	125.1	125.1	113.	112.7	112.8	112.3	117.9	117.7	104.5
Heating surface, tubes, square feet	1161.4	1252.6	1422.7	1489.	1643.	1549.	1579.	1491.	2024.3
Heating surface, total, square feet	1286.5	1377.7	1535.7	1601.7	1755.8	1661.3	1696.9	1608.7	2128.8
Diameter of boiler inside largest ring, inches	49.6		49.7	55.1		55.1	59.1		59.1
Area of grate, square feet	25.2		24.1	25.5		23.9	23.5		22.4
Diameter of driving wheels, inches	78.74		78.74	59.06		59.06	49.6		49.6
Weight in working order on driving wheels, pounds	65,260		62,940	125,450		93,120	125,890		120,300
Weight in working order, total, pounds	117,950		111,340	125,450		114,060	125,890		120,500
Ratio of l. p. to h. p. cylinder	2.6		....	2.52		....	2.25		....
Ratio of receiver volume to one h. p. cylinder	5.58		....	4.24		....	3.79		....
Boiler pressure, gauge, pounds per sq. inch	213.3		156.4	213.3		142.2	213.3		156.4

198°, is approximately half way between 180° and 225°. For the freight locomotive the starting power was apparently given greater weight in the problem, the angles in each design being 225° *plus* the angle of inclination of the h. p. cylinders

Diagram of Turning Moment at Driving Axle
Compound Locomotive, Lindner System.

FIG. 166.

L. Diagram of Turning Moments of a Lindner Two-Cylinder Receiver Compound, 103–107.—Fig. 166 shows an interesting diagram devised by Lindner to explain the turning moment of the Lindner compound at the driving axle. It applies to a Lindnei express engine in which the h. p. crank leads. The boiler pressure is 180 pounds by gauge. The cylinder ratio is 2 2.

The path of the crank and the periphery *a* of the wheel are shown on a scale of $\frac{1}{30}$ natural size, and the tangential forces transmitted to the periphery of the wheel are shown as lines *b* extending outward from the circumference, each millimeter of their departure from *a* representing one kilogram.

The adhesion circle *c* corresponds to a tractive force of 4,000 kilograms equal $\frac{1}{7}$ of the adhesion weight, while the mean tractive force calculated from the dimensions of the locomotive according to the formula $z = \frac{0.5 \times d^2 l p}{D}$ (which is the formula for compound locomotives) is given at 3,700 kilograms, d and l representing the diameter and stroke respectively of the h. p. cylinder, p the boiler pressure, and D the diameter of the driving wheel.

For the conditions of starting with various positions of crank, four phases or periods must be considered for each semi-circle.

(1) Phases II–III and VI–VII:

Within these phases the impulse takes place with increasing force from the h. p. piston alone without any back pressure, and as in every ordinary locomotive.

(2) Phases IV–V and VIII–I:

The impulse takes place with a force which increases within these phases, and is produced by the l. p. cylinder alone, with at least the same force as in phases II–III and VI–VII, while the h. p. piston is at the same time relieved by the small equalizing channels in the high-pressure slide,

(3) Phases I–II and V–VI:

The impulse is derived mainly from the l. p. piston acting with a great force, which, however, diminishes within the phases, while the h. p. piston acts with small but increasing force, and the motive effect of these two forces are combined.

(4) Phases III–IV and VII–VIII:

The impulse, where the auxiliary steam is led from the regulator, is derived solely from the h. p. piston with a great force, which diminishes somewhat within the phases, as represented by the pressure line *b l*.

Where the auxiliary steam is led from the main steam pipe, the l. p. piston also works simultaneously with small but increasing force. The pressure which is then developed on the l. p. piston acts back upon the h. p. piston; but, since the h. p. crank is in the most favorable position, the impulse is speedily given by the h. p. piston, and the motive effect does not in any case fall below the minimum values at the commencement of the phases indicated by (1) and (2) where only one piston is working at a time, but will correspond with the pressure line.

M. Some Tests of Compound Locomotives in the United States.

TABLE II.

Giving List of some Tests of Compound Locomotives which have been made in the United States, and References.

Type of Compound.	Where Tested.	Reference.	Comparative Value of Test.
Vauclain	Baltimore & Ohio R.R.	R. R. G., 1890—Pages 627, 634, 651, 668, 711	Fair
Johnstone	Mexican Central Ry.	" 1891 " 350, 352, 354	"
Vauclain.	Pennsylvania R.R ..	" " " 363	Unimpr't.
Vauclain	Lehigh Valley R.R	" " " 363	"
Johnstone..... .	Mexican Central Ry..	" " " 373	"
Vauclain & Dean	Lehigh Valley R.R.	" " " 655	"
Johnstone	Mexican Central Ry..	" " " 775	"
Johnstone	Mexican Central Ry .	" " " 812	"
Vauclain	Mexican National Ry.	" " " 858	"
Johnstone	Mexican Central Ry	" 1892 " 113	"
Vauclain.	West. Maryland R.R.	" " " 195	"
Dean	Old Colony R.R.....	" " " 384	"
Vauclain	Illinois Central R.R.	" " " 442, 443, 444.........	Fair
Vauclain	C., M. & St. P.	" " " 471, 472-5, 488, 490-94	"
Vauclain	C., M. & St. P..	" " " 577, 583-4	"
Johnstone	Mexican Central Ry .	" " " 631-3...............	"
Dean	Old Colony R.R.....	" " " 708	Unimpr't.
Vauclain	M. K. & T. R.R. ..	" " " 838	"
Pitkin..........	E. Tenn., Virg. & Ga.	" " " 987	"
Vauclain	Cin., N. O. & Tex.Pac.	" 1893 " 162, 163, 170........	"
Vauclain	N. Y., Chic. & St. L..	" " " 312	"
Lindner	C., B. & Q...	" " " 202-6, 211, 273......	Fair
Lindner	C., B. & Q..	" " " 313, 314, 335-7-8....	"
Vauclain	C., B. & Q.........	" " " 202-6, 211..........	"
Dean	Old Colony R.R......	" " " 273, 462	"
Vauclain	West. N. Y. & Penn.	Ry. Review, 1891 " 657	"
Pitkin	E. Tenn., Virg. & Ga.	Evg.News, Nov. 1890, Pages 458...........	Unimpr't.
Pitkin.	E. Tenn., Virg. & Ga.	" " Dec. 1891, " 545-6.........	"
Rhode Island. ..	Union Elev., Brooklyn	" " Dec. 1890, " 565...........	Fair
Rhode Island....	Union Elev., Brooklyn	" " Jan. 1891, " 6-10........	"
Rhode Island....	Union Elev., Brooklyn	" " Dec. " " 556-7.........	"
Pitkin.	Southern Pacific. ..	" " Feb. " " 193...........	"
Vauclain.......	Central R.R. of N. J..	" " Jun. 1892, " 588...........	Unimpr't.
Vauclain.......	C., M. & St. P..... ..	" " Jun. " " 636-9, 646-8, 657	Fair
Vauclain.	C., M. & St. P... ...	" " Jul. " " 6	"

N. Some Reported Savings by Compounds in the United States:

TABLE H H.

Giving some of the Reported Savings of Compound Locomotives in the United States.

NOTE.—This data is generally unreliable, as it was not accurately taken. Therefore, no attempt has been made to draw conclusions or make averages.

Railroad Company.	Type of Compound.	Builder.	Service.	Reported Saving in Fuel per cent.	Reported Saving in Water per cent.
Mexican Central	4 cyl. Johnstone (No. 66)	Mexican Central R. R. Co.	Freight	25.0	15.0
Cornwall and Lebanon	4 " Vauclain	Baldwin Locomotive Works	Freight	24.5	....
Western R. R. of Cuba	4 " Vauclain	Baldwin Locomotive Works	Freight	25.5	....
East Tennessee, Virginia & Georgia	2 " Pitkin	Schenectady Loco. Works	Passenger	24.7	....
East Tennessee, Virginia & Georgia	2 " Pitkin	Schenectady Loco. Works	Freight	22.4	21.8
Chicago & South Side	4 " Vauclain	Baldwin Loco. Works	Elevated Rapid Transit	30.0	....
Chicago, Burlington & Quincy	2 " Lindner	C., B. & Q. R. R.	Freight	20.0	....
Mexican Central Ry	4 " Johnstone	Rhode Island Loco. Works	Freight	22.0	....
Northern Pacific R. R.	4 " Vauclain	Baldwin Loco. Works	Freight	22.2	11.3
Western New York & Penn. R. R.	4 " Vauclain	Baldwin Loco. Works	Freight	36.2	17.9
Norfolk & Western R. R.	4 " Vauclain	Baldwin Loco. Works	Freight	38.	38.
Chicago, Milwaukee & St. Paul	4 " Vauclain	Baldwin Loco. Works	Freight	22.3	14.1
Union Elevated, Brooklyn	2 " Batchellor	Rhode Island Loco. Works	Elevated Rapid Transit	37.7	....
Illinois Central Suburban	4 " Vauclain	Baldwin Loco. Works	Passenger	17.	....
Western Maryland	4 " Vauclain	Baldwin Loco. Works	Freight	44.9	....
Old Colony R. R	2 " Dean	Old Colony R. R. shops	Freight	31.9	13.3
Old Colony R. R	2 " Dean	Old Colony R. R. shops	Passenger	29.6	....
Old Colony R. R	2 " Dean	Old Colony R. R. shops	Passenger	30.0	31.4
Missouri, Kansas & Texas	4 " Vauclain	Baldwin Loco. Works	Freight	28.7	23.3
Cin., New Orleans & Texas Pacific	4 " Vauclain	Baldwin Loco. Works	Freight	35.5	28.7
New York, Chicago & St. Louis	4 " Vauclain	Baldwin Loco. Works	Freight	29.3	....
Norfolk & Southern	4 " Vauclain	Baldwin Loco. Works	Freight	22.2	....
Norfolk & Southern	4 " Vauclain	Baldwin Loco. Works	Passenger	7.0	....
New York, Lake Erie & Western	4 " Vauclain	Baldwin Loco. Works	Freight	8.0	....
Chicago & Northwestern	2 " Pitkin	Schenectady Loco. Works	Freight	7.7	....
Chicago & Northwestern	4 " Vauclain	Baldwin Locomotive Works	Passenger	10.0	....
Kings County Elevated	2 " Batchellor	Rhode Island Loco. Works	Rapid Transit	24.0	....
Cleveland, Akron & Columbus	4 " Vauclain	Baldwin Loco. Works	Passenger	15.0	....
Mexican Central	4 " Johnstone	....	....	20.0	....
Norfolk & Western	4 " Vauclain	Baldwin Loco. Works	Passenger	13.4	8.6
Chicago, Milwaukee & St. Paul	4 " Vauclain	Baldwin Loco. Works	Freight	10 7	....
Pennsylvania & Northwestern	4 " Vauclain	Baldwin Loco. Works	Freight	18.0	....
Southern Pacific	2 " Pitkin	Schenectady Loco. Works	Freight	8.2	....
Cincinnati Southern	4 " Vauclain	Baldwin Loco. Works	Passenger	25.0	....
Vandalia Line	2 " Colvin	Pittsburgh Loco. Works	Freight	24.	15.5
Chesapeake & Ohio	2 " Mellin	Richmond Loco. Works	Freight	23.9	....
Mexican National	4 " Vauclain	Baldwin Loco. Works	Freight	17.8	....

O. Formulas for Expansion Curve.—The formula for the rectangular hyperbola is

$$P' V' = C'$$

in which P' is the absolute pressure at any point of the stroke, V' the total volume occupied by the steam, and C' a constant number depending upon the pressure and volume at cut-off, or at any point in the stroke that is used as the basis for comparison. To find C' take the volume occupied by the steam at any point of the stroke and multiply it by the absolute steam pressure at that point. To find the pressure at any point in the stroke divide C' by the total volume at that point.

The formula for the adiabatic expansion curve is approximately

$$P' V'^{\frac{10}{9}} = C'$$

In this formula the letters have the same signification as given above, and the value of C' is found in the same way. To find the pressure at any point divide C' by the total volume at that point raised to $\frac{10}{9}$ power. The powers and roots are best obtained by means of logarithms. To find the 10th power, take the logarithm of the number and multiply it by 10, then find the number corresponding to this product in the logarithmic table. To extract the 9th root of a number, take the logarithm of the number and divide by 9, and find the number in the logarithmic table corresponding to the quotient. The adiabatic curve of expansion is one that takes into consideration the amount of steam condensed in doing useful work in the cylinders, as distinguished from that condensed by the cooling effect of the cylinder walls. The hyperbola does not allow for condensation, but simply assumes an expansion where the pressure varies inversely as the volume occupied by the steam; that is, when the volume is made twice, three or four times as large, the pressures would be ½, ⅓, and ¼ respectively.

The saturation curve or curve of equal steam weight can be plotted directly from a steam table, which gives the volume of an equal weight of steam at different absolute pressures.

The curve of equal total heat can be plotted from a steam table which gives the total heat of an equal weight of steam at different absolute pressures.

P. Formula for Inertia of Reciprocating Parts.—Especially in high speed engines the inertia of the reciprocating parts materially alters the distribution of the pressure on the crank pin during the stroke, although the mean effective pressure as shown by the indicator card for each stroke is not changed. The only effect of the inertia of reciprocating parts is to reduce the pressure on the crank pin during the first part of the stroke and increase it during the last part of the stroke. During the first half of the stroke the velocity of the reciprocating parts is increased, and during the last half the velocity is decreased. In the beginning of the stroke a portion of the power of the steam is used to accelerate the reciprocating parts, and in the latter part of the stroke the pressure on the crank pin is increased by the force required to retard the reciprocating parts.

The simple formula for the inertia of the reciprocating parts at any angle of the crank a is:

$$\frac{.031\ \mathrm{W}\ \mathrm{V}^2}{\mathrm{R}}\ \text{Cosine a.}$$

This formula does not take into account the obliquity of the connecting rod, but it is quite near enough for ordinary analysis to omit this factor, particularly where the connecting rod has considerable length in proportion to the stroke. The shorter the connecting rod the more necessary it is to include the obliquity of the rod and the formula for such cases can be found in technical books on steam engines.

In the foregoing formula W is the weight in pounds of the piston, piston rod, crosshead and part of the connecting rod. The portion of the connecting rod to be taken varies with the design. For all ordinary analyses take ½ of the weight of the rod.

R = radius of crank in feet.

V = equals the velocity of the crank pin in its circular path around the axle. This velocity may be found by multiplying the velocity of the train in feet per seconds by the stroke of the cylinders and dividing by the diameter of the drivers.

a is the angle of the crank with the horizontal line through the wheel centres at the point where it is desired to find the inertia of reciprocating parts. The cosine of the angle may be found from any book giving a table of *natural* sines and cosines as distinguished from the *logarithmic* sines and cosines.

The inertia of the reciprocating parts is to be subtracted from the total steam pressure on the piston for the first half of the stroke and added to the total steam pressure on the piston for the last half of the stroke in order to find the actual pressure on the crank pin.

The following is an example of the application of this formula:

The weight of reciprocating parts 600 pounds = W.

Velocity of train 60 miles an hour or 88 feet per second

Diameter of drivers 6 feet.

Stroke of piston 2 feet.

Total piston pressure 38170 pounds.

Angle of crank with horizontal line through centres of drivers $=35° = a$.

Position of crank = first half of stroke.

Cosine of $35° = .819$.

Velocity of crank pin in circular path $= \frac{88 \times 2}{6} = 29.3 = V$

The square of 29.3 is $858. = V^2$

The inertia of the reciprocating parts at an angle of 35° is

$$\frac{.031 \times 600 \times 858.}{1} \times .819 = 12988 \text{ pounds.}$$

The actual pressure on the crank pin, when the crank has moved 35 degrees from the end of the stroke, is the difference between the total steam pressure and the inertia of the reciprocating parts, or—

$$38170 - 12988 = 25182 \text{ pounds.}$$

Q.

TABLE L.

Comparative Cylinder Capacities of Compound Locomotives.

By whom Operated or Built.	Type of Engine.	Diameter of l. p. Cylinder. Inches.	Diameter of h. p. Cylinder. Inches.	Length of Stroke.	Diameter of Drivers. Inches.	Weight on Drivers. Tons.	Volume of l. p. piston displacement per foot of track per ton on drivers. A comparative figure.	Freight or Express.	Remarks.
Saxon State R. R.	Lindner	25.6	18.1	24	55.6	29.0	58.8	Frgt.	2 Cylinder.
" " "	"	23.6	16.5	22.1	75	32.0	30.8	Exp.	2 "
" " "	"	25.6	16.5	22.1	75	32.0	36.2	"	2 "
C. B. & Q.	"	29	20	24	68	45.8	38.8	"	2 "
Vladikavkaz	"	28	$19\frac{5}{8}$	$25\frac{9}{16}$	$47\frac{1}{4}$	49.9	50.8	Frgt.	2 "
Pennsylvania R. R.	"	31	19.5	28	84	42.0	46.1	Exp.	2 "
Michigan Central R. R.	Schenectady	29	12	24	68	48.5	36.8	"	10 Wheel 2 Cyl.
" " "	"	29	20	24	74	49.5	32.8	"	10 " 2 "
Southern Pacific	"	28	20	26	55	49.8	44.6	Frgt.	10 " 2 "
" "	"	29	20	24	69	48.3	36.5	Exp.	10 " 2 "
Adirondack & St. Lawrn'e	"	30	20	26	57	57 3	43.0	Frgt.	Mogul 2 "
" " "	"	30	20	26	70	54.0	37.3	Exp.	10 Wheel 2 "
Pennsylvania R. R.	"	30	20	24	74	53 0	33.2	"	10 " 2 "
East Tenn. Va. & Ga.	"	29	20	24	51	57.8	41.1	Frgt.	Consol. 2 Cyl.
Brooklyn "L"	Rhode Island	18	$11\frac{1}{2}$	16	42	15.8	46.8	Pass.	Forney 2 Cyl.
N. Y., N. H. & H. R. R.	"	28	18	24	78	33.3	43.6	Exp.	8 Wheel 2 Cyl.
Minneapolis & St. Louis	"	28	19	24	68	33.5	49.7	"	8 " 2 "
Northern Adirondack	"	28	19	26	62	46.4	42.8	"	10 " 2 "
Fitchburg	"	31	21	26	63	54.0	44.2	Frgt.	Mogul 2 "
M. St. P. & Ste. Marie.	"	31	21	24	50	59.1	47.0	"	Consol. 2 Cyl.
N. Y., N. H. & H. R. R.	"	31	21	26	78	42.0	46.1	Exp.	8 Wheel 2 "
Chicago M. & St. Paul	"	31	21	26	78	45.0	43.0	"	10 " 2 "
Grafton & Upton	"	28	18	24	55	43.0	47.8	Frgt.	Mogul Forney 2"
Lake Street Elevated	"	21	13	18	44	21.0	51.8	Pass.	8 Wheel 2 Cyl.
Northeast England	Worsdell	28	20	24	$91\frac{1}{4}$	18.5	66.6	Exp.	2 Drivers, 2 Cyl.
Old Colony R. R.	Dean	28	20	24	69	33.2	49.0	"	8 Wheel 2 "
Lehigh Valley R. R.	"	30	20	24	50	28.7	90.8	Frgt.	Consol. 2 "
C. & S. S. R. T. R. R.	Baldwin	20	14	16	42	20.0	45.9	Pass.	Forney 2 "
Lake Shore & Mich. So.	Brooks	$28\frac{1}{2}$	18	24	56	38.3	52.6	Frgt.	10 Wheel 2 "
" " " "	Cooke	27	19	24	64	48.5	34.0	"	10 " 2 "
Jura-Simplon	Mallet	26.4	17.7	25.6	72	30.6	48.8	Exp.	American 2 "
" "	Pittsburgh	29	19	26	72	—	—	"	8 Wheel 2 "
" "	"	29	19	26	54	47.5	51.5	Frgt.	Mogul 2 "
Chesapeake & Ohio	Richmond	29	19	24	57	—	—	"	10 Wheel 2 "
C. C. C. & St. L.	"	30	19	24	56	—	—	"	10 "
Illinois Central R. R.	Rogers	29	20	26	56	53.7	43.8	"	Mogul 2 "
West India Imp. Co.	"	29	20	26	50	48.5	54.3	"	10 Wheel 2 "
Jura, Berne-Lucerne	von Borries	25.5	18.5	26	59	—	—	Exp.	8 " 2 "
Pennsylvania R. R.	Webb	30	14	24	75	33.4	51.6	"	3 Cylinder.
London & N. W.	"	30	15	24	85	34.7	44.0	"	8 Wheel 3 Cyl.
Chemnitz, Eng. Works	Lindner	$18\frac{1}{2}$	$11\frac{7}{8}$	21	$43\frac{1}{4}$	33.0	60.6	Frgt.	4 Cylinder.
Baltimore & Ohio	Vauclain	20	12	24	66	37.8	46.0	"	8 Wheel 4 Cyl.
Northern Pacific	"	19	11	24	56	45.0	41.2	"	Consol. 4 "
Bahia Exten. Brazil	"	15	9	18	37	26.5	49.6	"	4 Cylinder.
New South Wales	"	22	$13\frac{1}{2}$	26	51	61.5	48.0	Exp.	10 Wheel 4 Cyl.
West. N. Y. & Penn.	"	21	13	26	$50\frac{1}{4}$	58.2	47.2	Frgt.	10 " 4 "
C. & S. S. R. T. R. R.	"	15	9	16	42	20.0	51.8	Exp.	Forney 4 "
Cent. R. R. of Georgia	"	19	$11\frac{1}{2}$	24	68	30.0	51.	"	8 Wheel 4 "
Cen. R. R. of New Jersey	"	22	13	24	78	44.2	40.4	"	8 " 4 "
Columbian Exposition	"	22	13	26	$84\frac{1}{4}$	41.6	43.4	"	4 "
" "	"	24	14	24	72	46.8	49.6	"	10 Wheel 4 "
Missouri, Kansas & Texas	"	24	14	26	56	67.0	48.0	Frgt.	Consol. 4 "
N. Y. L. E. & W.	"	27	16	28	50	85.0	58.0	"	Decapod 4 "
Paris, Lyons & Med.	4 Cylinder	19.7	12.2	24.4	78.7	32.5	44.4	Exp.	

TABLE L.—*Continued.*

By whom Operated or Built.	Type of Engine.	Diameter of l. p. Cylinder. Inches.	Diameter of h. p. Cylinder. Inches.	Length of Stroke.	Diameter of Drivers. Inches.	Weight on Drivers. Tons.	Volume of l. p. piston displacement per foot of track per ton on drivers. A comparative figures.	Freight or Express.	Remarks.
Paris, Lyons & Med.	4 Cylinder	21.3	13.4	25.6	59 0	62.5	38.0	Frgt.	
" " "	4 "	21.3	14.2	25.6	49.5	63.0	44.4	"	
Decauville	Mallet articulated	11.0	7.4	10.2	23.6	12.9	48.4	"	4 Cylinder. 4 "
Herault	"	18 1	12.1	20.5	47.2	38.5	44.0	"	4 "
Central Suisse	"	21.7	14.0	25.2	55.1	65.0	40.0	"	4 "
Gothard	"	22.8	15.8	25.2	48.4	93.7	34.4	"	4 "
Alsatian Constructors	Woolf	20.9	13.4	25 2	83.2	33.6	47.6	Exp.	8 Wheel 4 Cyl.
No. R. R. of France	"	26.0	15.0	25.6	51.2	56.9	71.2	Frgt.	Tandem 4 Cyl.
Great Northern	Brooks	22.0	13.0	26	55.0	65.0	42.6	"	" Consol.
So. W. Russia	Mallet	19.7	13.0	23.6	79.0	93.7	70.9	Exp.	"
Hungarian State	Woolf	19.2	13.6	26	78.5	30.8	48.0	"	"
Mexican Central	Johnstone	24¼ 3 to 1	14.0	24	48.0	50.0	70.4	Frgt.	Annular Cyls. 4 Cyls.
" "	"	24¼ 3 to 1	14.0	24	56.0	52.5	57.6	"	Annular Cyls. 4 Cyls.
" Northern	"	24¼	14.0	24	56.0	51.5	59.2	"	4 Cylinders
Mex. Cuernavaca & Pac.	"	24¼	14.0	24	56.0	51.5	59.2	"	4 "
Mexican Central	"	22½	13.0	24	48.0	50.0	61.0	"	4 "
" "	"	22½	13.0	24	56.0	38.0	68.8	Exp.	4 "
" "	"	22½	13.0	24	48.0	50.0	61.0	Frgt.	4 "
" "	"	22½	13.0	24	48.0	105.0	58.4	"	Double Bogie.

R. TABLE C C.

Dimensions of some of the more Prominent Compound Locomotives that have been put into Actual Service, Chiefly in the United States.

Reference No.	Builder.	Patentee.	Type of Compound System.	Railroad Company.
1	Baldwin Locomotive Works	S. M. Vauclain	4 Cyl.	C. & S. S. R. T. R. R.
2	" " "	" "	4 "	Central R. R. of Ga.
3	" " "	" "	4 "	Central R. R. of N. J.
4	" " "	" "	4 "	Columbian Exposition
5	" " "	" "	4 "	" "
6	" " "	" "	4 "	Missouri, Kan. & Tex.
7	" " "	" "	4 "	N. Y., L. E. & W.
8	" " "	————	2 "	C. & S. S. R. T. R. R.
9	Neustadt Locomotive Works	C. Gölsdorf	2 "	Nothrn Ry of Austria.
10	Northern R'y of France	————	3 "	" " " France.
11	Brooks' Locomotive Works	J. Player	2 "	Lake S. & Mich. So.
12	" " "	" "	Tandem	Great Northern.
13	Chicago Burlington & Quincy	A. Lindner	2 Cyl.	C., B. & Q.
14	Chemnitz Engine Works	" "	4 "	Royal Saxon State.
15	Pennsylvania R. R. Shops	" "	2 "	Pennsylvania R. R.
16	Cooke Locomotive Works	————	2 "	Experimental.
17	Kolomna Engine Works (Moscow)	von Borries	2 "	St. Petersb. & Warsaw.
18	Old Colony R. R.	F. W. Dean	2 "	Old Colony.
19	Lehigh Valley R. R.	" "	2 "	Lehigh Valley.
20	Alsatian Constructors	A. Mallet	Tandem	South West Russia.
21	J. A. Maffie, Munich	" "	4 Cyl.	St. Gothard.
22	" " "	" "	4 "	Central Suisse.
23	————	" "	2 "	Jura Simplon
24	Alsatian Constructors	————	4 "	So. R. R. of France.
25	Hungarian Ry. Shops, Buda-Pesth	————	Tandem	Hungarian State.
26	So. Eastern Ry, Gateshead Shops	T. W. Worsdell	2 Cyl.	North Eastern.
27	Richmond Locomotive Works	C. J. Mellin	2 "	Chesapeake & Ohio.
28	" " "	" "	2 "	C. C. C. & St. L.
29	Rhode Island Locomotive Works	F. W. Johnstone	4 "	Mexican Central.
30	" " " "	" "	4 "	" Northern.
31	" " " "	" "	4 "	M. Cuernavaca & Pac.
32	" " " "	" "	4 "	Mexican Central.
33	" " " "	" "	4 "	" "
34	" " " "	C. H. Batchellor	2 "	Brooklyn Union "L."
35	" " " "	" "	2 "	N. Y., N. H. & H.
36	" " " "	" "	2 "	Minneap's & St. Louis.
37	" " " "	" "	2 "	Northern Adirondack.
38	" " " "	" "	2 "	Fitchburg.
39	" " " "	" "	2 "	M. St. P. S. St. M.
40	" " " "	" "	2 "	N. Y., N. H. & H.
41	" " " "	" "	2 "	C. M. & St. P.
42	" " " "	" "	2 "	Grafton & Upton.
43	" " " "	" "	2 "	Lake St. "L." Chicago
44	Mexican Central R. R.	F. W. Johnstone	4 "	Mexican Central.
45	" " "	" "	4 "	" "
46	Rogers Locomotive Works	————	2 "	Illinois Central.
47	" " "	————	2 "	West India Imp. Co.
48	Schenectady Locomotive Works	A. J. Pitkin	2 "	Southern Pacific.
49	" " "	" "	2 "	Adirondack & St. L.
50	" " "	" "	2 "	" "
51	" " "	" "	2 "	Pennsylvania.
52	" " "	" "	2 "	E. Tenn., Va. & Ga.
53	" " "	" "	2 "	Michigan Central.
54	London & North Western	F. W. Webb	3 "	London & N. W.
55	" " "	" "	3 "	" " "
56	" " "	" "	3 "	" " "
57	" " "	" "	3 "	" " "
58	" " "	" "	3 "	" " "
59	" " "	" "	3 "	" " "
60	" " "	" "	3 "	" " "
61	Hanover Machine Works	A. von Borries	2 "	Prussian State.
62	————	" " "	2 "	Grand Trunk.
63	Neilson & Co., Glasgow	Worsdell & von Borries	2 "	Bengal & Nagpur, Ind.
64	————	A. von Borries	2 "	Jura, Berne-Lucerne.

TABLE C C.—*Continued.*

Reference No.	Type of Engine.	Service for which Engine was built.	Diameter and Stroke of Cylinders. Inches.	Diameter of Drivers. Inches.	Weight on Drivers. Pounds.
1	Forney	Elevated	9 & 15 × 16	42	40,000
2	8 Wheel	Passenger	11½ & 19 × 24	68	60,000
3	"	High Speed Pass.	13 & 22 × 24	78	88,400
4	Special High Speed	" " "	13 & 22 × 26	84¼	83,140
5	10 Wheel	" " "	14 & 24 × 24	72	93,580
6	Consolidation	Freight	14 & 24 × 26	56	134,100
7	Decapod	"	16 & 27 × 28	50	170,000
8	Forney	Elevated	14 & 20 × 16	42	40,000
9	6 Wheel	Freight	19¾ & 29¼ × 25	50¾	——
10	Mogul	"	17 & 19.7 × 27.6	65	90,940
11	10 Wheel	Fast Freight	18 & 28½ × 24	56	76,500
12	Consolidation	Freight	13 & 22 × 26	55	130,000
13	Mogul	"	20 & 29 × 24	62	97,000
14	Double Bogie	"	11⅞ & 18⅛ × 21	43¼	——
15	8 Wheel	Passenger	19.5 & 31 × 28	84	84,000
16	10 "	Freight	19 & 27 × 24	64	97,000
17	8 "	Passenger	18 & 26 × 25.5	78	52,000
18	8 "	Passenger & Freight	20 & 28 × 24	69	66,000
19	Consolidation	Freight	20 & 30 × 24	50	109,088
20	8 Wheel	Passenger	13 & 19.7 × 23.6	79	57,300
21	Articulated	Freight	15.8 & 22.8 × 25.2	48.4	187,300
22	"	"	14 & 21.7 × 25.2	55.1	130,000
23	8 Wheel	Passenger	17.7 & 26.4 × 25.6	72	61,270
24	8 "	"	13.4 & 20.9 × 25.2	83.2	67,240
25	8 "	"	13 & 19.2 × 26	78.5	61,508
26	8 "	"	20 & 28 × 24	91¼	39,760
27	10 "	Freight	19 & 29 × 24	57	89,000
28	10 "	"	19 & 30 × 24	56	107,100
29	10 "	"	14 & 24½ × 24	56	103,000
30	10 "	"	14 & 24½ × 24	56	103,000
31	10 "	"	14 & 24½ × 24	56	103,000
32	Double Bogie	"	13 & 22½ × 24	48	210,000
33	Consolidation	"	13 & 22½ × 24	48	100,000
34	Forney	Elevated	11½ & 18 × 16	42	31,534
35	8 Wheel	Fast Passenger	18 & 28 × 24	78	66,520
36	8 "	Passenger	19 & 28 × 24	68	66,950
37	10 "	"	19 & 28 × 26	62	92,880
38	Mogul	Freight	21 & 31 × 26	63	108,000
39	Consolidation	"	21 & 31 × 24	50	118,220
40	8 Wheel	Passenger	21 & 31 × 26	78	84,000
41	10 "	"	21 & 31 × 26	78	90,000
42	Mogul Forney	Freight	18 & 28 × 24	55	86,000
43	8 Wheel "	Elevated	13 & 21 × 28	44	43,000
44	10 "	Passenger	13 & 22½ × 24	56	76,000
45	Consolidation	Freight	13 & 22½ × 24	48	100,000
46	Mogul	"	20 & 29 × 26	56	107,300
47	10 Wheel	"	20 & 29 × 26	50	97,000
48	10 "	Passenger	20 & 29 × 24	69	96,680
49	Mogul	Freight	20 & 30 × 26	57	114,500
50	10 Wheel	Passenger	20 & 30 × 26	70	108,000
51	10 "	"	20 & 30 × 24	74	106,000
52	Consolidation	Freight	20 & 29 × 24	51	113,500
53	10 Wheel	Passenger	20 & 29 × 24	74	99,000
54	6 "	"	13 & 26 × 24	81	61,264
55	6 "	"	14 & 30 × 24	75	67,200
56	6 "	"	14 & 30 × 24	85	69,440
57	8 "	"	15 & 30 × 24	85	69,440
58	Side Tank	"	14 & 26 × 24	56½	69,664
59	" "	"	14 & 26 × 24	68½	65,632
60	" "	Freight	14 & 30 × 24	62½	64,512
61	6 Wheel	Passenger	16½ & 23⅝ × 22⅞	73¼	28,672
62	——	——	18 & 28.5 × 26	——	——
63	10 Wheel	Freight	18 & 26 × 26	51	——
64	8 "	Passenger	18.5 & 25.5 × 26	59	——

TABLE C C.—*Continued.*

Reference No.	Area of Grate. sq. ft.	Heating Surface Fire box. sq. ft.	Total Heating Surface. sq. ft.	Remarks.
1	19.0	70.0	554.5	Nos. 1 to 45.
2	17.6	140.44	1567.07	"Nancy Hanks."
3	38.5	174	1711.0	No. 385.
4	24.6	128.23	1478.13	"Columbia."
5	18.7	152.5	1793.0	"Columbus."
6	25.1	164.3	1768.0	No. 231.
7	89.6	234.3	2443.1	Nos. 800 to 805.
8	19.0	70.0	555.0	No. 46.
9	——	——	——	One of Five Engines.
10	22.5	——	1224.9	Built in 1888.
11	23	112	1298	
12	25.3	117	1596	
13	27.1	126.0	1506	No. 324, Design of Mr. William Forsyth.
14	15	59	930	Meyer-Lindner Duplex.
15	30.0	159	1820	Class T, Design of Mr. Axel S. Vogt.
16	28.0	——	1766.6	No. 1.
17	26.5	134.6	1572	"One of Nineteen."
18	19.2	——	1354	
19	69	——	1658	No. 310.
20	20.4	111.9	1317	
21	23.7	100.1	1661.1	12 Driving Wheel "Goliath."
22	19.4	86 1	1345.6	8 " " Double Bogie.
23	21.6	——	1390.5	With "von Borries" Starting Gear.
24	22	117	1211.5	Designed by Mr. Du Bousquet.
25	32.3	129.1	1451.5	Duplex.
26	20.7	123	1139.0	No. 1518.
27	——	——	——	No. 140.
28	——	——	——	
29	27.3	204	2004	Six Engines.
30	27.3	204	2004	One Engine.
31	28.3	213 4	2013.4	" "
32	43.4	274.0	2570	Three Engines.
33	21.5	148	1348	One Engine.
34	16.5	56.5	299.5	
35	18.5	138.0	1229	
36	23.5	156.0	1425	
37	29.0	165.5	1469.5	
38	27.0	172	1493	
39	25.0	160	1700	
40	34.5	166.5	1395.5	
41	27.0	204.0	1788.0	
42	17.5	91.0	831.0	
43	17.1	57.6	537.5	
44	17.0	122	1222	No. 66, Altered from Simple Engine.
45	21.5	148	1348	" " " "
46	26.5	156	1516	
47	26.5	164	1621	
48	29.3	——	1736.2	No. 1785.
49	——	——	——	No. 50.
50	——	——	——	No. 15.
51	26.2	141.7	1953.2	No. 1503.
52	——	——	——	No. 461.
53	28.2	141.2	1992.6	No. 338, for "North Shore Limited."
54	17.1	103.5	1063.7	"Experiment."
55	20.5	159.1	1379.6	"Dreadnaught."
56	20.5	159.1	1401.5	"Teutonic."
57	20.5	120.6	1505.7	"Greater Britain."
58	14.24	84.8	993.6	
59	14.24	84.8	993.6	
60	17.1	94.6	1098.8	
61	18.7	78.5	1047.8	Built in 1885.
62	——	——	——	
63	22	104	1223.0	Built in 1887.
64	16	80.5	1302	

GLOSSARY.

A.

Absolute Pressure.—Gauge pressure plus 14.7 pounds.

Actual Cut-Off.—The cut-off which includes a consideration of the clearance; the quotient of the volume of the cylinder at the cut-off point including the clearance, divided by the total volume of the cylinder including the clearance on one end.

Actual Indicator Cards.—Cards taken from actual engines as distinguished from elementary cards drawn according to the elementary theory of steam engines.

Angularity of Connecting Rod.—The angle which the connecting rod makes with the line through the centre of the cylinder at any point during a revolution.

Apparent Cut-Off.—The cut-off shown by the indicator card; the cut-off measured from the valve motion; a cut-off that does not take into account the clearance in the cylinders.

Atmospheric Line.—The line of no pressure as shown by steam gauge; a line drawn at 14.7 pounds above the line of zero of absolute pressures.

B.

Back Pressure.—The pressure in the cylinders against the piston on the return stroke; the pressure against which the piston is moving.

By-Pass Valve.—A valve which, when opened, permits the steam to pass from one end of a cylinder to the other.

C.

Clearance.—The volume into which the steam left in the cylinder, when the exhaust port is shut, is compressed; the cubical contents of the space between the piston, when at the end of its stroke, and the face of the valve seat, including all ports and connecting passages and indicator pipes if any.

Combined Indicator Card.—A diagram showing the cards from both cylinders drawn to the same scale of volumes and pressures.

Compression.—Reduction of volume of the steam enclosed in the cylinder after the exhaust opening is shut; the opposite of expansion.

Continuous Expansion.—Expansion that goes on without interruption as in a single expansion engine; the Woolf type; the Vauclain type; the Johnstone and the DuBousquet; expansion without pause, as in the case of receiver engines where steam pauses in the receiver between the two expansions, viz., one in the h. p. and one in the l. p. cylinder.

Cut-Off.—The point where steam is shut off from admission to the cylinders; the point of the stroke where expansion begins.

E.

Elementary Compound.—A compound engine that is assumed to give an elementary indicator card; an engine assumed for the purpose of discussion and illustration.

Elementary Indicator Cards.—Cards that do not take into account the losses of pressure and volume in actual engines; sometimes called theoretical indicator cards.

Elementary Theory.—Limited theory; theory that does not take into consideration a majority of the practical conditions as distinguished from the more perfect or complete theory.

Expansion Curve.—A curve which shows the variation of pressure during expansion.

I.

Inertia of Reciprocating Parts.—The tendency of reciprocating parts to remain at rest or at a constant velocity; the inertia is measured by the force required to get the reciprocating parts up to speed or to reduce the speed or to stop them.

Initial Condensation.—Condensation which takes place before cut-off.

Initial Pressure.—Pressure at the beginning of the stroke.

Inside Clearance—Negative Lap.—The opening of the steam port to the exhaust cavity of the valve when the valve is at its centre of motion.

Intercepting Valve.—The valve which prevents the steam, admitted from the boiler to the l. p. steam chest, from passing through the receiver to the h. p. cylinder.

L.

Link Motion.—All of the distributing apparatus such as eccentrics, links, etc.; frequently intended to include the valve and other parts affecting the control of the steam pressure in the cylinders.

M.

Mean Forward Pressure.—The average pressure on the piston which pushes it forward.

N.

Negative Lap.—See Inside Clearance.

Non-Receiver Engines.—Compound engines without receivers; continuous expansion engines; the Woolf, the Vauclain, the Johnstone and the DuBousquet.

O.

Outside Lap.—The distance which the steam valve laps over the steam port when the valve is at its centre of motion.

P.

Potential of Pressure.—The amount of pressure; the pressure above the atmosphere; the intensity of pressure; used to emphasize the fact that wire-drawing causes a loss of potential or force; strictly, the term is equivalent to pressure.

R.

Ratio of Cylinders.—Ratio of cylinder volumes, not including clearance; where the stroke is the same for both cylinders it is the ratio of cylinder areas.

Ratio of Expansion.—The ratio of the initial pressure to the final pressure in the cylinder; the quotient of the initial pressure divided by the final pressure; sometimes taken as the quotient of the final volume divided by the volume at cut-off, clearance being included.

Re-Admission.—Admission of steam the second time during a stroke; increase of steam pressure during admission to l. p. cylinder caused by exhaust from h. p. cylinder.

Receiver Engine.—A compound with a receptacle or receiver for the steam exhausted from the h. p. cylinder; not a continuous expansion engine.

Reciprocating Parts.—The parts that move forward and back and do not revolve; piston, piston rod, crosshead and part of connecting rod.

Re-Evaporation.—The evaporation of the initial condensation; the evaporation of moisture in the steam.

Release.—The point where the exhaust valve opens; the end of expansion.

S.

Sequence of Cranks.—The location of cranks with respect to each other in rotation.

Single Expansion.—The expansion of steam in one cylinder; not compound.

Steam Use.—Transforming the heat in steam into mechanical work; utilization of steam in cylinders; method of using steam.

Super-Heating.—The heating of steam above the temperature which it normally has at the same pressure in a steam boiler; steam can only be super-heated when separated from water.

T.

Tandem.—Cylinders placed one in front of the other, *i, e.*, placed in tandem.

Total Expansion.—The ratio of the initial pressure in the h. p. cylinder to the final pressure in the l. p. cylinder.

V.

Valve Gear.—All of the valve motion which regulates the distribution of steam in the cylinders.

Valve Motion.—See Valve Gear.

W.

Wire-Drawing.—Throttling steam through an aperture; a reduction of pressure by restricting the flow of steam; drawing through a small opening.

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